HEAT TRANSFER IN EXTENDED SURFACE EQUIPMENT WITH VARIABLE REFRIGERANT TEMPERATURE

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by

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Professor J. S. Newell
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Dear Sir:

In accordance with the requirements for the Degree of Naval Engineer, we submit herewith a thesis entitled, "Heat Transfer in Extended Surface Equipment with Variable Refrigerant Temperature."

ACKNOWLEDGMENT

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TABLE I

Symbols and Units Used.

A_1 Area of duct upstream of nozzle, sq. ft.

A_2 Area of nozzle throat, sq. ft.

A_f Face area of test section, sq. ft.

A_i Inside cross-sectional area of finned tube, sq. ft.

B Ratio of outside surface to inside surface of finned tube.

C Nozzle discharge coefficient.

C_{pa} Specific heat of dry air, Btu/lb. - deg. F.

C_{ps} Specific heat of superheated water vapor Btu/lb. - deg. F

C_{pm} Humid specific heat, Btu/lb. - deg. F

D_1 Mean diameter of duct upstream of nozzle, ft.

D_2 Diameter of nozzle throat, ft.

D_i Inside diameter of finned tube, ft.

f_R Film coefficient of heat transfer between the internal surface of the tube and the fluid flowing inside.
Btu/hr. - sq. ft. (internal surface) - deg. F

f_a Film coefficient of heat transfer between the external surface of the tube and the air flowing outside.
Btu/hr. - sq. ft. (external surface) - deg. F

Note: Subscript o, $(f_R)_o$, denotes observed coefficient; subscript c, $(f_R)_c$, enotes computed coefficient.

h_{a1} Enthalpy of air and attendant water vapor at entrance to test section, Btu/lb.

h_{a2} Enthalpy of air and attendant water vapor at exit from test section, Btu/lb.

h_{fc} Enthalpy of condensed water from tube surface corresponding to t_c, Btu/lb.

h_{f1} Enthalpy of cooling water at tube inlet, Btu/lb.

h_{f2} Enthalpy of cooling water at tube outlet, Btu/lb.

P_{aN} Partial pressure of dry air in duct at upstream side of nozzle, lbs./sq. in.

P_b Barometric pressure, lbs./sq. in.

P_N Static pressure at nozzle inlet, lbs./sq. in.

P_{sN} Partial pressure of water vapor in duct at upstream side of nozzle, lbs./sq. in.

Δp_N Static pressure drop across nozzle, lbs./sq. in.

Q_a Rate of dry air flow through nozzle, cu.ft./min.

Q_{as} Rate of flow of dry air at standard conditions, cu.ft./min. (70 deg. F, 29.92 ins. Hg. Bar.)

Q_m Rate of flow of mixture through nozzle, cu.ft./min. (Note: $Q_a = Q_m$)

q_a Heat removed from air by cooling water Btu/hr.

q_w Heat transferred to cooling water, Btu/hr.

R_a Gas constant for dry air = 53.35 ft. lb./lb. - deg. F.

Re Reynold's Number

S_o Outside surface of finned tube, sq. ft.

S_i Inside surface of finned tube, sq. ft.

t_{dp} Dew point of entering air, deg. F

t_N Dry bulb temperature at nozzle, deg. F

T_N Air temperature at nozzle, deg. F absolute.

t_i Dry bulb temperature at inlet to plenum chamber, deg. F.

t_i' Wet bulb temperature at inlet to plenum chamber, deg. F.

t_1 Dry bulb temperature at inlet to test section, deg. F.

t_1' Wet bulb temperature at inlet to test section, deg. F.

Δt Dry bulb temperature drop across test section, deg. F.

t_2 Dry bulb temperature at outlet of test section, deg. F.

t_2' Wet bulb temperature at outlet of test section, deg. F.

t_o Dry bulb temperature at exhaust from duct, deg. F.

t_o' Wet bulb temperature at exhaust from duct, deg. F.

t_c Temperature of condensate from tube, deg. F.

t_s Tube surface temperature, deg. F.
(t_{s1} at inlet, t_{s5} at outlet, t_{s2}, t_{s3} and t_{s4}, intermediate surface temperatures).

t_{w1} Cooling water inlet temperature, deg. F.

t_{w2} Cooling water outlet temperature, deg. F.

$t_{w\,av}$ Average cooling water temperature, deg. F.

$(t_s' - t_w)_m$ Mean temperature difference between tube surface and water, deg. F.

V_f Face velocity of dry air, ft./min.

V_{fs} Face velocity of dry air at standard conditions, ft./min.

V_w Velocity of water through tube, ft./sec.

v_{aN} Specific volume of dry air at nozzle inlet, cu.ft./lb.

v_{mN} Specific volume of mixture of dry air and water vapor at nozzle inlet, cu.ft./lb.

v_{std} Specific volume of dry air under standard conditions of temperature and pressure = 13.37 cu. ft./lb.

w_a Rate of flow of dry air, lbs./min.

w_m Rate of flow of mixture, lbs./min.

w_w Rate of flow of cooling water, lbs./min.

W Specific humidity, lbs. water vapor per lb. dry air.

W_1 Specific humidity at inlet to test section.

W_2 Specific humidity at outlet of test section.

ν Kinematic viscosity of water, sq.ft./sec.

γ_a Density of dry air, lbs./cu. ft.

γ_s Density of water vapor, lbs./cu. ft.

γ_m Density of mixture, lbs./cu. ft.

γ_{std} Density of dry air at standard conditions = 0.075 $\frac{\text{lbs.}}{\text{cu.ft.}}$

Σ Sigma function, Btu/lb. dry air.
(subscript "1" refers to entering and "2" to exit conditions of test section).

CHAPTER I

SUMMARY

OBJECT: This investigation into the heat transfer involved in the use of chilled water to cool air had a two-fold purpose: to investigate the behavior of the refrigerant film coefficient at different rates of water and air flow and under different air and water conditions; and to investigate the feasibility of using a small test set-up to achieve results which could be extended to a full coil.

METHOD: A test set-up as nearly in accord with A.S.R.E. Test Codes as possible was designed. Sufficient data were obtained in the fifteen thirty-minute runs to determine the state of the air at each end of the test section and the temperature of the water at inlet and outlet. The water rate was determined by a weigh barrel and the air flow by a standard nozzle and standard A.S.M.E. discharge coefficient curve. The tube surface temperatures were obtained by imbedded thermocouples and the temperature drop of the air across the test section was obtained by six thermocouples in series.

The original plan of work called for using one, two, and three finned tubes and cooling the inlet water; the time limit prevented fulfillment of this plan. Tap water was used throughout with no cooling apparatus; runs were made with one finned tube only.

RESULTS: From the quantitative results of Table II, Chapter IV, the following formulas were developed:

$$f_R = 290\,(V_w)^{0.44}$$

$$f_R = 555\left(\frac{60\,V_w}{V_{fs}}\right)^{0.40}$$

Both formulas were obtained from log-log plots.

As a supplementary result of this experiment, a graphical means is presented in Appendix E of determining the specific volume of dry air, which simplifies the calculations involved in finding the mass rate of flow of dry air in a duct.

CONCLUSIONS: Water velocity is unquestionably the greatest factor in determining the refrigerant film coefficient. The above empirical formulas disagree with those presented in the literature to a sufficient extent to warrant further investigation. The experimental procedure appears to be satisfactory for determining film coefficients but more precision in temperature measurements is necessary.

RECOMMENDATIONS: Further investigation should be carried out using multiple tubes and a greater range of water temperatures. A larger blower would permit runs to be made at approximately 250, 500, and 750 feet per minute standard air velocity. The installation of thermocouples in series in the inlet and outlet water ends and use of a more precise potentiometer would permit a check between the heat transfer on the water and air sides. It is expected that agreement will be close. The nozzle should be calibrated with a pitot tube traverse to confirm accuracy of air flow measurements for future experimental work.

CHAPTER II

INTRODUCTION

The advent of extended surface cooling coils into the field of air conditioning complicated the problem of coil ratings and the calculation of performance after the ratings were established. A solution to this problem has been attempted primarily in two ways: by means of an overall coefficient of heat transfer and by means of individual surface film coefficients for air and refrigerant or surface temperature.

The first solution, commonly known as the "brute strength method," involves numerous tests and much data in the form of tables or curves for each specific coil that is to be rated. Although this method can be used to predict the performance of coils which are similar to the one tested, it lacks the flexibility of application to coils of different physical characteristics. The quantity of testing involved has caused the method to give way to the second approach using more exact heat transfer theory. Only a few tests are required to rate a coil over its entire operating range; calculations of performance are more difficult than with the direct method but the test results are more generally applicable to other coils.

Goodman (8) first used this approach in 1936 with an ingenious application of Lewis' so-called "straight line law." His procedure was to hold the refrigerant velocity constant so that the refrigerant film coefficient was

essentially constant; the air film coefficient varied as the air velocity raised to some power (f_a = const. X V^n). By measuring the overall coefficient of heat transfer and using McAdams' (4) graphical analysis, he found the necessary constant and exponent, hence the air film coefficient. Tuve and Seigel (9) (10) worked during the same period on a new theory, the "humidity method," which they presented in final form in 1945. Both air and refrigerant film coefficients were found by the same procedure that Goodman used for determining the air side coefficient: first holding the air side coefficient constant and allowing the refrigerant side coefficient to vary, then reversing the procedure.

In both of the above methods the emphasis was on the overall problem of calculating coil performance. The evaluation of the film coefficients, particularly on the refrigerant side, was only a means to the end. Because of the difficulty in obtaining a true surface temperature, measurement was made of the overall coefficient of heat transfer for use with McAdams' analysis to find the air side coefficient. Goodman presented no method for evaluating the refrigerant film coefficient; Tuve and Seigel assumed that it varied as the refrigerant velocity raised to the 0.8 power, with a different constant for each type of coil. In the ASHVE Guide (5), an empirical formula relates the refrigerant film coefficient to refrigerant velocity raised to the 0.8 power but allows the constant to vary slightly with average refrigerant temperature.

A variation in the value of the refrigerant film coefficient can have a great effect on the accuracy of coil performance calculations; the published range was considered too broad to be correct. In view of the lack of reported experimental results, this work was undertaken to examine the behavior of this coefficient under various conditions. One object, therefore, of this experimental work was to supplement and corroborate the existing data for this important air conditioning application of the use of water in air cooling. Another object was to evaluate the accuracy of the results from a small test set-up such as was used in the experimental work.

Every effort has been made to maintain the symbols and units of Table I throughout the thesis report. The one necessary exception to uniformity is in Table IV, Appendix C, which has an explanatory note, Figures V, VI and VII of Appendix A show details of the equipment and are the plans from which the major units were fabricated by the Boston Naval Shipyard. Figures VIII, IX and X, Appendix A, are photographs of the equipment as mounted in the laboratory for test runs.

The Aerofin coil was selected as a typical example of an extended surface tube used in air conditioning applications. The physical properties of the coil are summarized in Table III of Appendix A. In working with one tube instead of with a coil composed of several lengths of tube, many precautions were necessary to reduce heat losses which might amount to a large percentage of the heat transfer involved

in the test runs. Two typical precautions were the aluminum foil insulation on the test section to reduce conduction and radiation and the mounting of the tube in fiberboard to prevent any contact between the tube and duct walls.

It is hoped that the accuracy of the results and the comparative ease of procedure will make the methods used in the experimental work of value in determining performance characteristics of extended surface tubes and coils. The qualitative results and recommendations of Chapters V, VI, and VII will perhaps be of sufficient merit, in any case, to extend the knowledge of the subject of cooling with a variable refrigerant temperature.

CHAPTER III

PROCEDURE

A series of runs was made at as great a variation in inlet air conditions, air velocity and water velocity as the equipment would permit. The runs were usually of 30 minutes' duration with data recorded every five minutes. These data are summarized in Tables IV and V of Appendix C. The location of all measuring equipment is shown in Figure I and details are shown in Figure XI of Appendix B.

All temperatures were measured by copper-constantan thermocouples, static pressure and pressure drop across the nozzle by inclined Ellison draft gages with a range of one inch of water, the flow of water by a weigh tank and the flow of air by the static pressure drop across a standard nozzle. By means of the aspirating psychrometer at the plenum inlet, the specific humidity at the equipment inlet, hence at the test section inlet, was determined. This fact plus the dry bulb temperature obtained from the thermocouple located at the test section inlet determined the state of the air at the inlet to the test section. The differential thermocouple across the test section accurately measured the temperature drop in the air stream, and permitted simple computation of the dry bulb temperature at the test section outlet. The state of the air leaving the test section was determined by the dry bulb temperature and specific humidity at that point. This specific humidity was the same as that of the air leaving the equipment and was obtained by an aspirating psychrometer

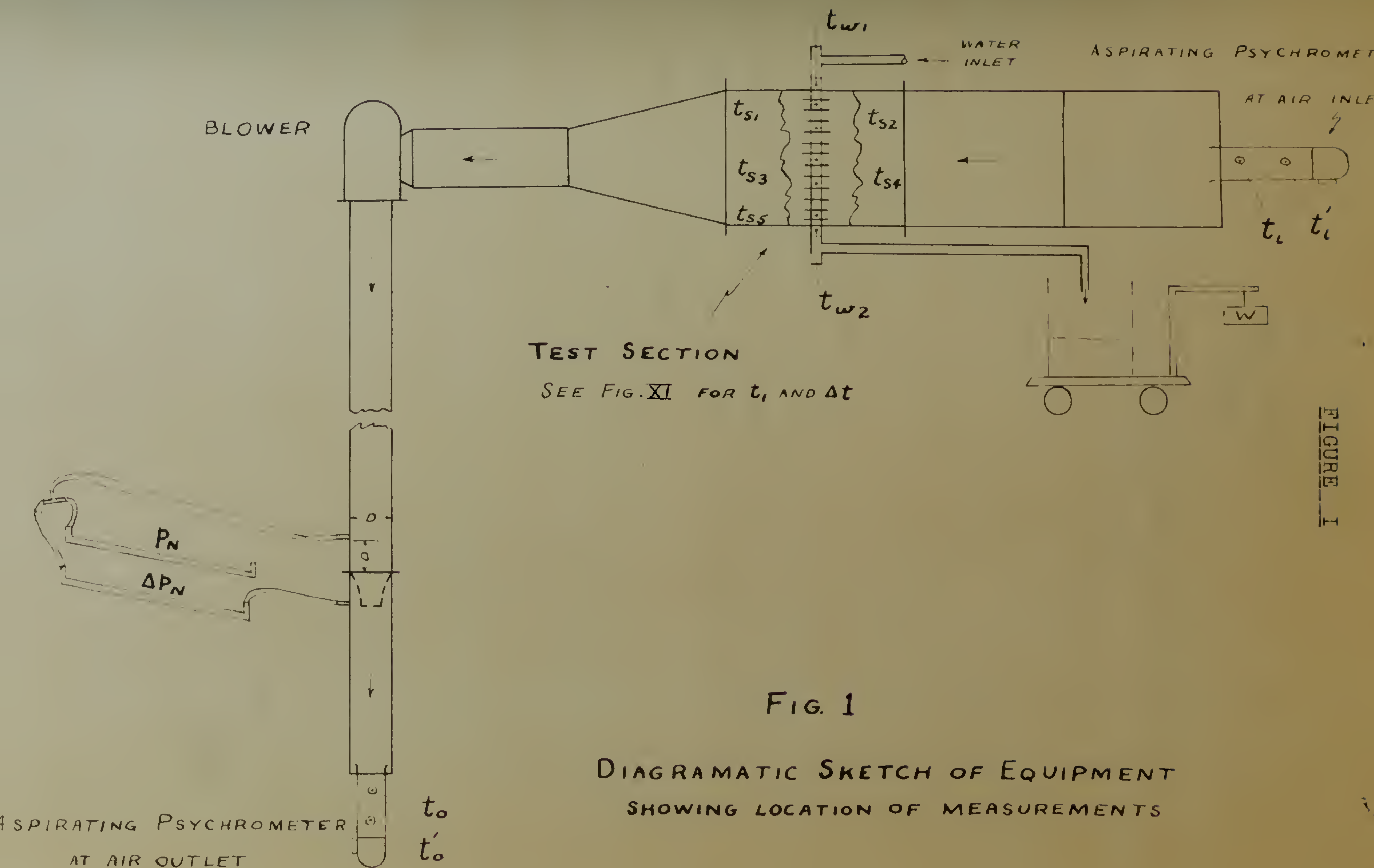
at the equipment outlet.

Because of the lack of time and extreme difficulty in placing thermocouples on the inner surface of a small diameter finned tube, the temperature drop through the tube wall was assumed to be negligible. This assumption is usually made in developing formulas for coil performance and in obtaining the film coefficients in thin-walled copper tubes. The five surface temperature thermocouples were imbedded in the tube surface in slots at equal distances along the length of the tube from inlet to outlet side.

The temperature of the air at the upstream side of the nozzle was assumed to be the same as at the outlet of the equipment. A small variation would have little effect in computing the specific volume of the dry air and of the mixture of dry air and water vapor. In the tabulated results of Chapter IV mass rates of flow of dry air and of the mixture are given; volumetric rate of flow and face velocity are tabulated for standard air conditions, 70 degrees Fahrenheit and 29.92 inches of mercury barometer.

Necessary deviation plots of the thermocouples used, the A.S.M.E. discharge coefficient curve for the standard nozzle and a table of kinematic viscosities are included in Appendix C. The method of computing the results is indicated by the sample calculations of Appendix D. This same appendix contains the list of formulas used in the computations.

FIGURE I



TEST SECTION

SEE FIG. XI FOR t_1 AND Δt

FIG. 1

DIAGRAMATIC SKETCH OF EQUIPMENT

SHOWING LOCATION OF MEASUREMENTS

CHAPTER IV

RESULTS

The results obtained from calculations using the observed data are presented in Table II.

Figure II shows the plot of the computed values of refrigerant film coefficient vs. water velocity for various values of the average water temperature. The empirical formula is indicated on the plot. The dash line is the plot of the observed value of the refrigerant film coefficient vs. the water velocity.

Figure III is a logarithmic plot of the log of the observed refrigerant film coefficient vs. the log of the water velocity. The formula derived from the plot is indicated on the Figure.

Figure IV is also a logarithmic plot showing the log of the regrigerant velocity vs. the log of the ratio of the water velocity to the face velocity of the air corrected to standard conditions. This plot is used to obtain the relationship shown on the Figure.

TABLE II
RESULTS

Run No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Date	Mar.22	Mar.25	Mar.25	Mar.29	Mar.29	Mar.30	Mar.30	Mar.31	Mar.31	Apr.5	Apr.5	Apr.5	Apr.5	Apr.8	Apr.8
Duration	45	45	30	30	30	30	30	30	15	30	30	30	30	30	30
Type of Run	Wet	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Dry	Wet	Wet	Wet	Wet	Dry	Dry
tDP	56.1	46.7	47.3	37.8	42.0	42.0	40.6	26.5	25.8	69.4	69.5	71.2	71.0	44.1	44.7
W1	0.0097	0.0070	0.0070	0.0049	0.0057	0.0057	0.0054	0.0030	0.0029	0.0154	0.0154	0.0165	0.0163	0.0062	0.0063
W2	0.0091	0.0083	0.0081	0.0070	0.0087	0.0057	0.0057	0.0031	0.0040	0.0149	0.0150	0.0160	0.0154	0.0068	0.0063
PsN	0.214	0.196	0.193	0.165	0.204	0.134	0.134	0.070	0.094	0.347	0.348	0.382	0.358	0.157	0.146
PaN	14.66	14.72	14.72	14.64	14.59	14.60	14.59	14.62	14.61	14.45	14.45	14.41	14.44	14.37	14.38
VaN	13.62	13.65	13.60	13.69	13.71	13.75	13.82	13.65	13.70	13.90	13.89	13.95	13.91	13.93	13.92
VmN	13.51	13.55	13.50	13.60	13.61	13.69	13.77	13.60	13.62	13.71	13.69	13.74	13.72	13.86	13.87
Qm = Qa	43.9	41.2	37.6	47.9	43.9	51.0	43.3	42.3	50.7	51.0	43.9	43.6	51.0	51.0	42.6
Wa	3.23	3.02	2.76	3.50	3.16	3.70	3.13	3.10	3.70	3.66	3.16	3.12	3.66	3.66	3.06
Wm	3.25	3.04	2.78	3.52	3.18	3.72	3.15	3.12	3.72	3.72	3.21	3.17	3.72	3.68	3.08
Qas	43.1	40.2	36.3	46.8	42.2	49.5	41.8	41.3	49.4	48.9	42.2	41.6	48.8	48.8	40.8
Vfs	299	279	256	325	293	344	290	288	343	340	293	290	339	338	284
qs	171.0	195.4	187.3	89.4	97.2	170.2	146.5	148.7	177.2	175.8	157.1	173.2	198.0	194.2	167.0
Ww	4.93	19.50	18.89	1.52	1.30	6.73	3.30	10.98	10.47	13.06	12.86	32.06	31.02	14.22	14.08
Vw	0.73	2.89	2.80	0.23	0.19	1.00	0.49	1.63	1.55	1.94	1.90	4.76	4.60	2.11	2.09
Re	2240	9065	8675	900	755	3404	1790	5374	5134	6680	6515	16057	15430	7138	7070
qw	See Chapter V, Discussion of Results, for explanation.														
(ts - tw)m	5.27	3.40	3.14	5.21	5.80	4.20	5.27	3.07	3.35	4.36	3.60	2.47	2.60	3.87	3.19
(fr)o	251	442	460	132	129	312	214	372	408	310	338	540	587	386	403
twav	43.57	44.89	44.17	59.34	60.44	50.22	54.68	48.05	48.41	50.93	50.66	49.62	49.24	49.80	49.80
(fr)c	187.1	568	550	81.4	72.0	253	146	367	354	429	422	868	845	456	453
% sensible heat	62.4	--	--	--	--	--	--	--	--	60.1	31.5	75.0	55.0	--	--

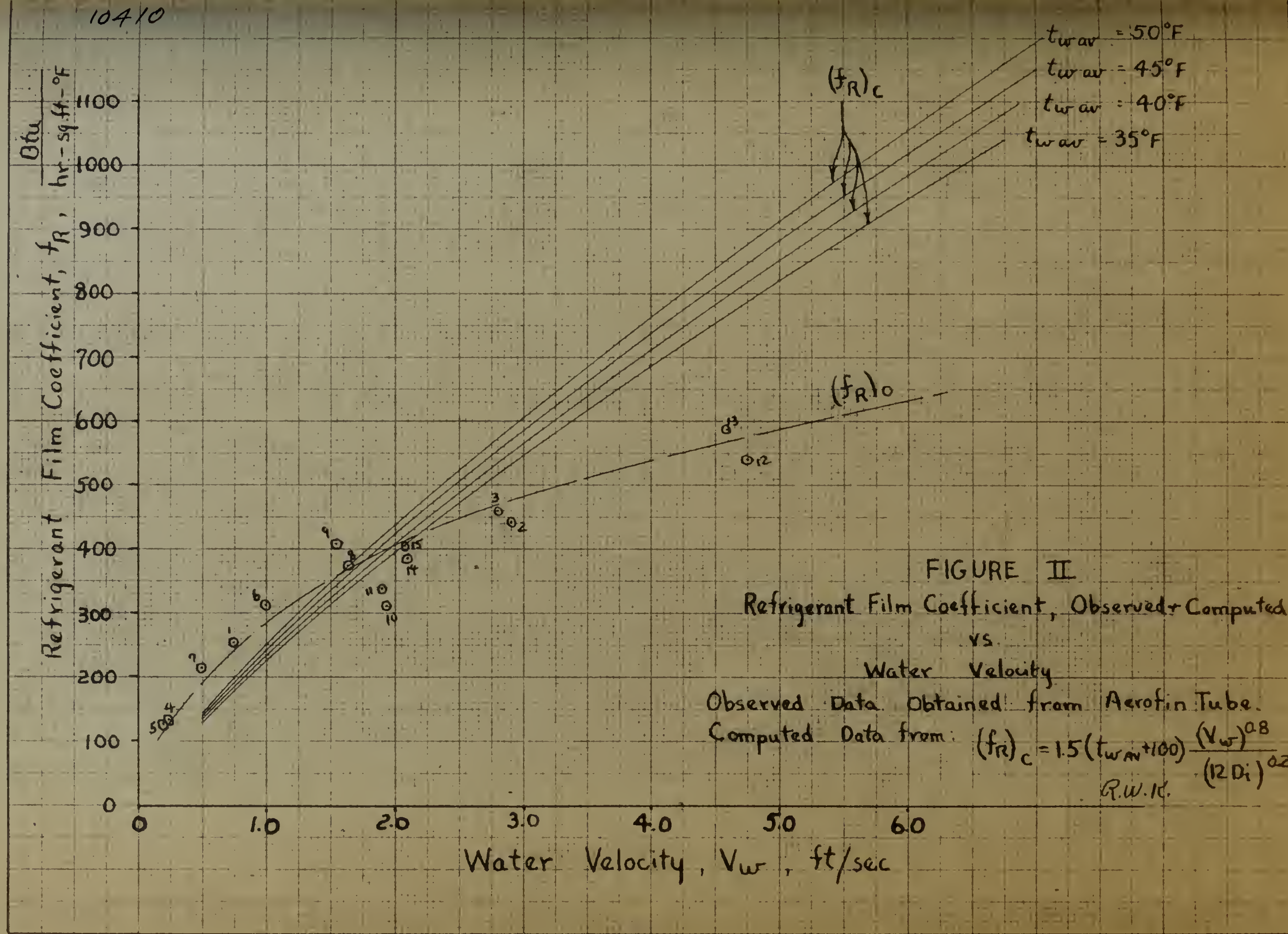


FIGURE II
Refrigerant Film Coefficient, Observed + Computed
vs
Water Velocity
Observed Data Obtained from Aerofin Tube.
Computed Data from: $(f_R)_c = 1.5(t_{w\,av}+100)\dfrac{(V_w)^{0.8}}{(12D_i)^{0.2}}$

FIGURE III

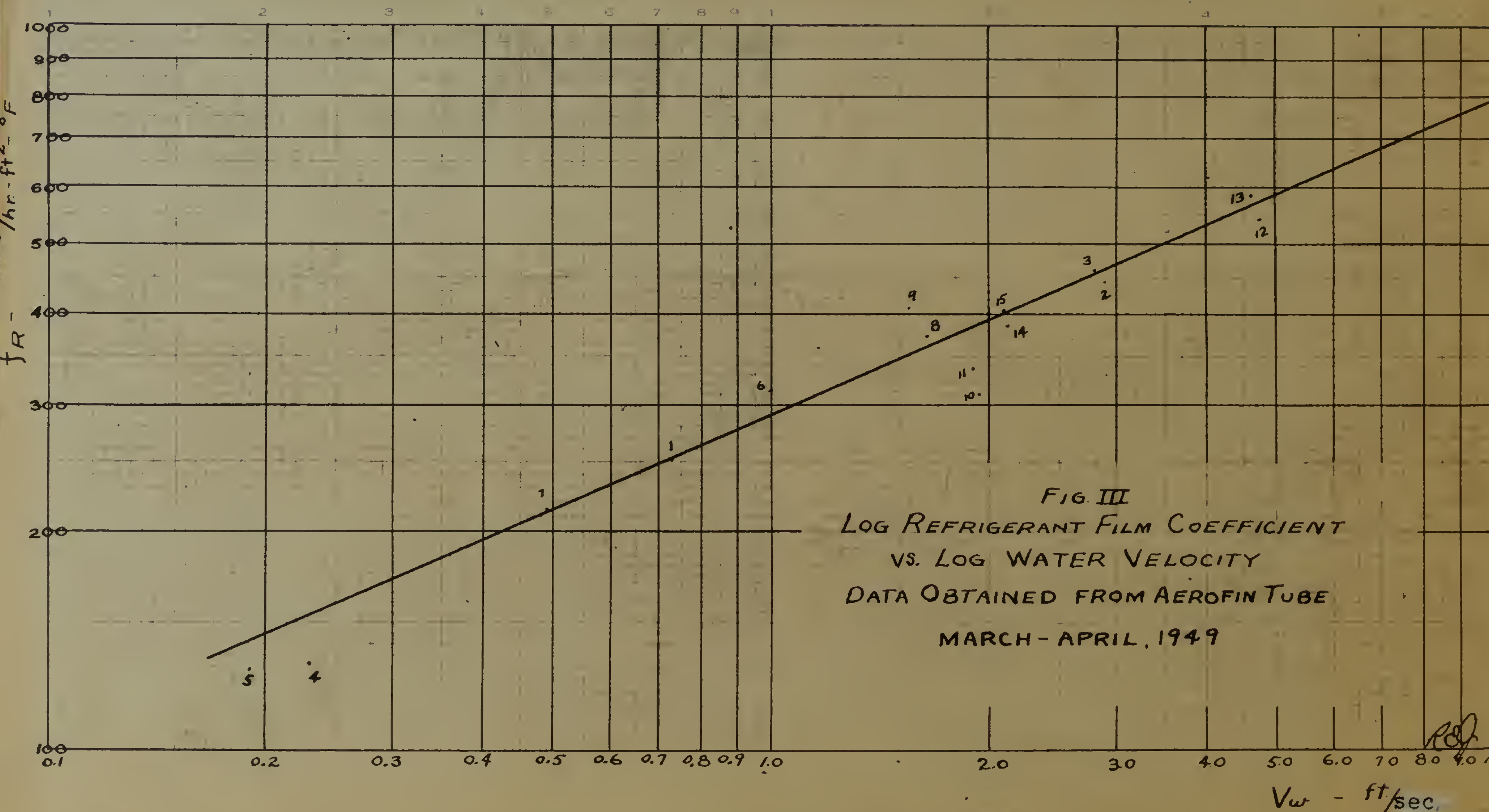
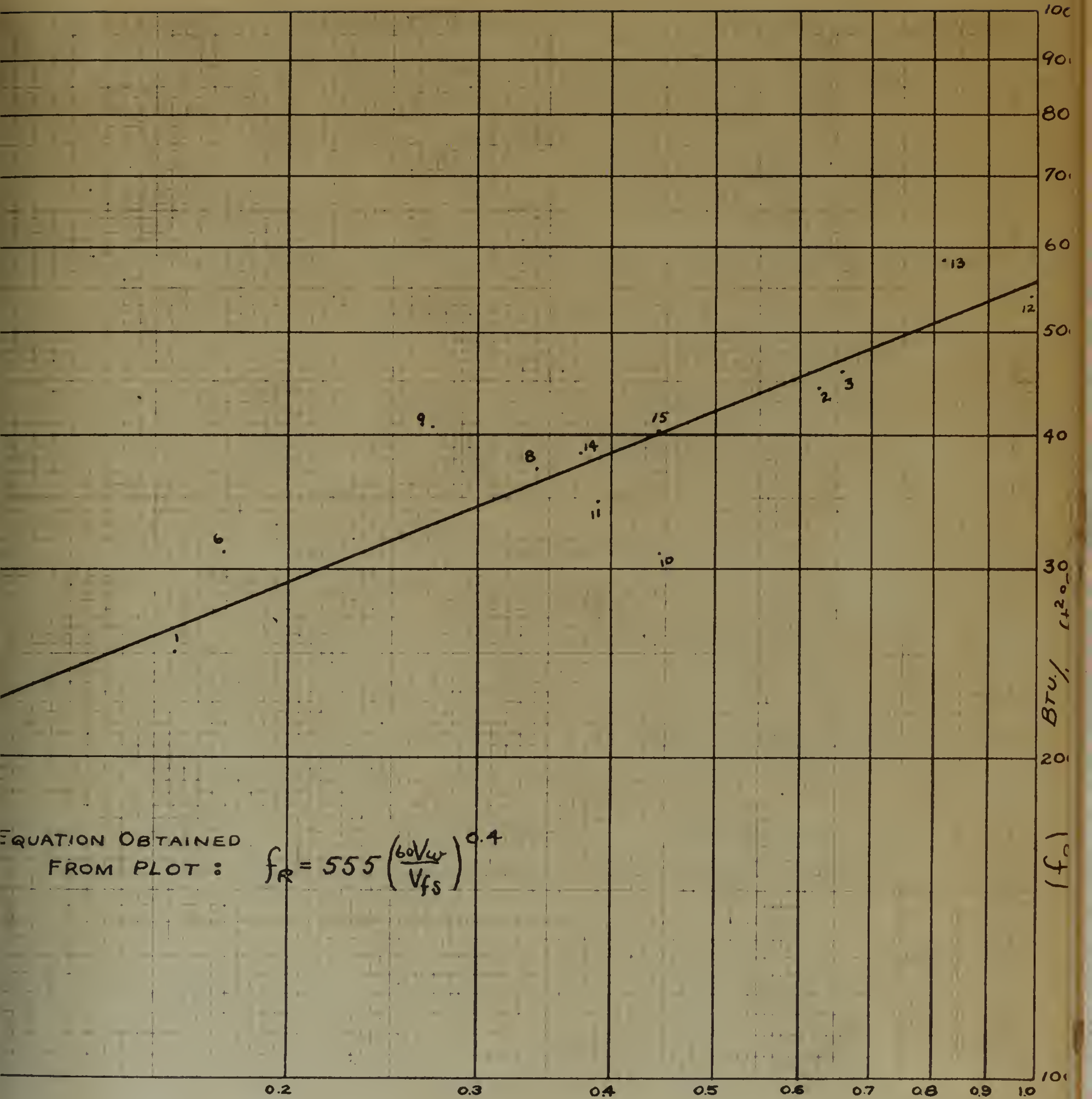


FIG. III
LOG REFRIGERANT FILM COEFFICIENT
VS. LOG WATER VELOCITY
DATA OBTAINED FROM AEROFIN TUBE
MARCH - APRIL, 1949

EQUATION OBTAINED FROM PLOT: $f_R = 290(V_w)^{0.44}$

FIGURE IV



LOG REFRIGERANT FILM COEFFICIENT VS.
LOG RATIO OF WATER VELOCITY TO FACE VELOCITY, STANDARD AIR
DATA OBTAINED FROM AEROFIN TUBE, MAR.-APR. 1949

KEY

CHAPTER V

DISCUSSION OF RESULTS

It is a common hypothesis that a film coefficient varies as the velocity of the fluid flowing over its surface; in this experiment the refrigerant film coefficient varies with the water velocity. Though the other variables were not negligible, the experimental results definitely showed that water velocity is the major independent variable. The formula as presented by the ASHVE Guide:

$$f_R = 1.5\left[t_{w\,av} + 100\right]\frac{(V_w)^{0.8}}{(12\,D_i)^{0.2}}$$

indicates that the film coefficient varies as the water velocity raised to the 0.8 power, multiplied by a constant which varies slightly with the size of tube and with temperature of the water. This relation was not confirmed by test results. The film coefficient calculated by this equation differed considerably from the observed coefficient over the normal operating range and was fifty percent or more in excess of the observed coefficient at higher water velocities. These discrepancies are shown by figure II.

From results of this experiment, it is suggested that a more accurate relation could be obtained from a function of water velocity to the 0.44 power:

$$f_R = 290\,(V_w)^{0.44} \qquad \text{FIG. III}$$

Film coefficients calculated from this formula are much more consistent with observed values not only over the normal operating range but beyond it at both extremes as was tested. Since some of the points in figure III were

rather dispersed, it was believed further that the refrigerant film coefficient might be affected by variations on the air side. Subsequently, a formula relating the film coefficient to the ratio of water velocity and standard air velocity was developed:

$$f_R = 555 \left(\frac{60V_w}{V_{fs}}\right)^{0.40} \qquad \text{FIG. IV}$$

The smooth curve obtained indicates that the refrigerant film coefficient cannot be declared wholly independent of varying conditions of the air side. This deduction may be justified in part by remembering that any heat transfer effects through the tube wall were neglected, or more precisely, were lumped together with the refrigerant film coefficient. Calculations made from this formula showed equally good agreement with observed data. Since the maximum air velocity obtainable, 340 feet per minute, included only the lower portion of the air-side operating range, the consistency of these relations should be tested at various values of air velocity up to 750 feet per minute. Though both developed formulas give much better results over a broader range of water velocities than the formula from the Guide for this one tube, it is recommended that they be compared with data observed from various arrangements of multiple coils before final acceptance.

The uniformity of results obtained from a wide variation of operating conditions justifies the adequacy of using a small scale setup for test work. To prevent an irregular by-pass factor, the face area for one tube was limited to

approximately the same amount as one tube would have in a full coil; and measures to prevent heat transfer losses were taken as noted in Chapter III. If similar precautions are used, it is believed that results obtained from a small scale test can be extended to a full coil with good accuracy.

One of the possible improvements for the calculations in this experiment is a calibration of the nozzle used to measure air flow as explained in appendix E. The area exposed to the water is small and the water temperature drop is very slight; it is recommended that multiple thermocouples be installed at the water inlet and outlet, and that a more precise potentiometer be used for the temperature measurements. This would furnish a check on the heat transfer as found from air side data. In this experiment a check was not possible since the water temperature drop was too small to be determined accurately by the potentiometer used. Since only tap water was available, the behavior of the film coefficient with water temperatures approaching freezing was not investigated. Temperature variations were found to be relatively insignificant but this should be confirmed for the lower temperature range also.

One discrepancy in this experiment was the inlet and exit air wet bulb temperature relationship which usually indicated an increase in specific humidity for the dry runs. After carefully shielding the psychrometers with aluminum foil to prevent radiation losses, a reasonable agreement was still lacking. This in no way detracts from

the results obtained. A careful check of the psychrometers with various thermocouples prior to further experimentation will probably show reason for discrepancy.

It is believed that adherence to the recommendations and precautions set forth will yield good results for fully analyzing the behavior of the refrigerant film coefficient over the entire range of possible operating conditions. The equipment can further be used to determine air side film coefficients and therefore the overall coefficient of heat transfer.

CHAPTER VI

CONCLUSIONS

1. Water velocity is the variable having the greatest effect on the refrigerant film coefficient.

2. The formula $f_R = 1.5\left[t_{w\,av} + 100\right]\dfrac{(V_w)^{0.8}}{(12\,D_i)^{0.2}}$ from the ASHVE Guide (5) appears to be inadequate.

3. The following formulas:

$$f_R = 290\,(V_w)^{0.44} \qquad \text{from Fig. III}$$

$$f_R = 555\left(\frac{60\,V_w}{V_{fs}}\right)^{0.40} \qquad \text{from Fig. IV}$$

were found to be more consistent with observed values. No preference was indicated as both have equal merit for the range of values covered by this experiment.

4. A small scale test setup, properly designed and used with care, will give reasonable results which can be extended to full size coils.

CHAPTER VII

RECOMMENDATIONS

1. The equipment should be designed with a blower capacity and speed control sufficient to reach an air velocity of 750 feet per minute in uniform steps.

2. The validity of the developed formulas should be confirmed by observed data from multiple coil arrangements.

3. The calibration of the nozzle used for measuring air flow should be checked by a pitot traverse for greater accuracy.

4. Multiple thermocouples in series should be used at water inlet and outlet.

5. A more precise potentiometer should be used for temperature measurements.

6. A water cooler should be included in the test equipment for lower water temperatures.

7. A thorough test of the aspirating psychrometers used should be made to insure identical operation and recording of temperatures under the same conditions.

APPENDIX

APPENDIX A

Supplementary Introduction

This section of the Appendix consists of Table III, the Properties of the Aerofin Tube used for the heat exchanger, Figures V, VI and VII, the plans from which the equipment was fabricated by the Boston Naval Shipyard, and Figures VIII, IX and X, photographs of the Test Equipment as set up in the Air-Conditioning Laboratory at M.I.T.

TABLE III

Physical Properties of Aerofin Helical-Type Fin Coil

Tube Data:

Material	-	Copper
O.D.	-	0.625 in.
I.D.	-	0.575 in.
Wall	-	0.025 in.
Inside section area	-	0.260 sq. in.

Fin Data:

Material	-	Copper
Fins per inch	-	7
Thickness	-	0.008 in.
Width (Helical)	-	0.375 in.
Type of Bond	-	Metallic

Surface (Sq. Ft.):

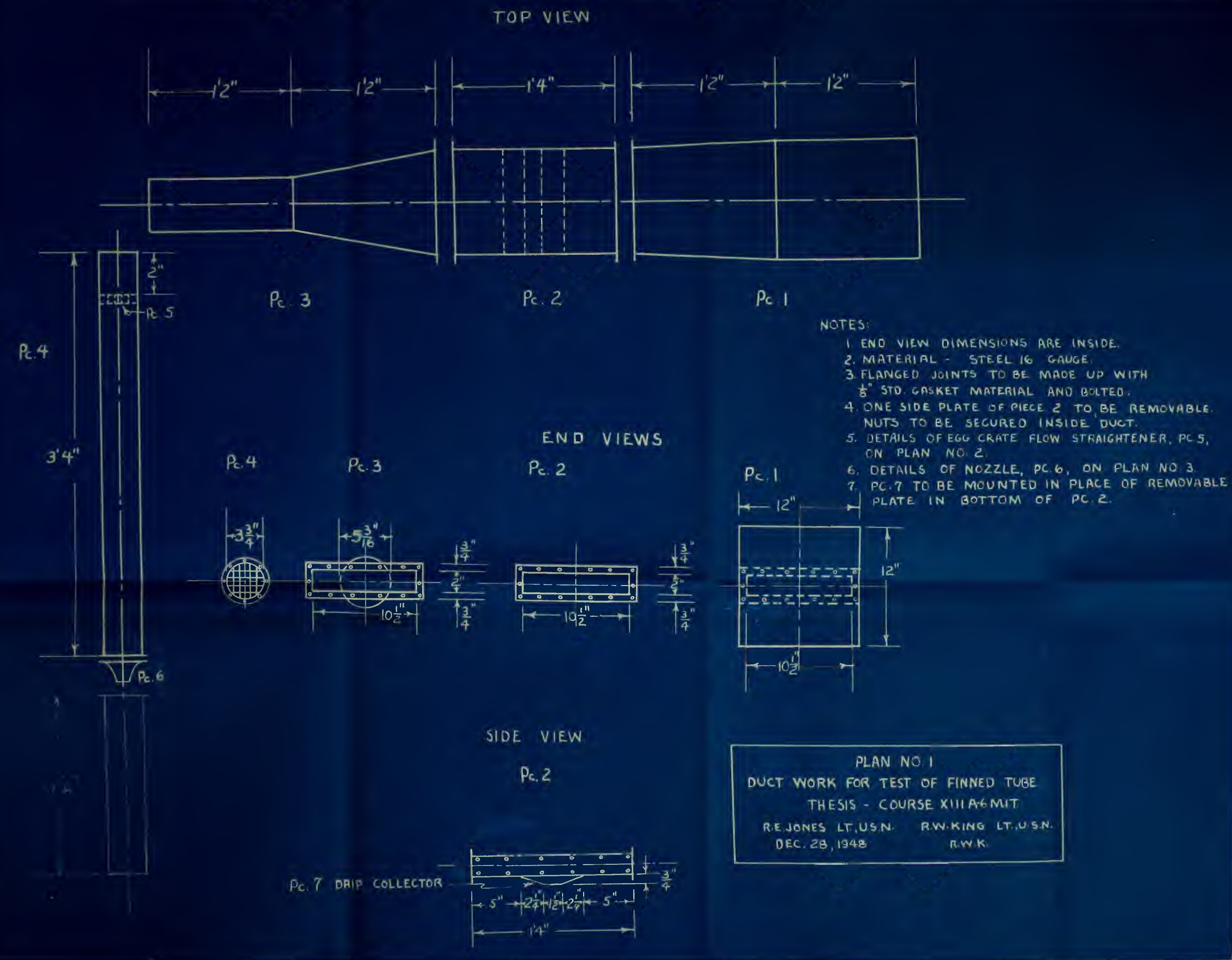
Inside per Linear Foot	-	0.1505
Outside per Linear Foot	-	1.50
Ratio Outside to Inside	-	10.0
* Outside per Sq. Ft. Face Area per Row	-	12.95

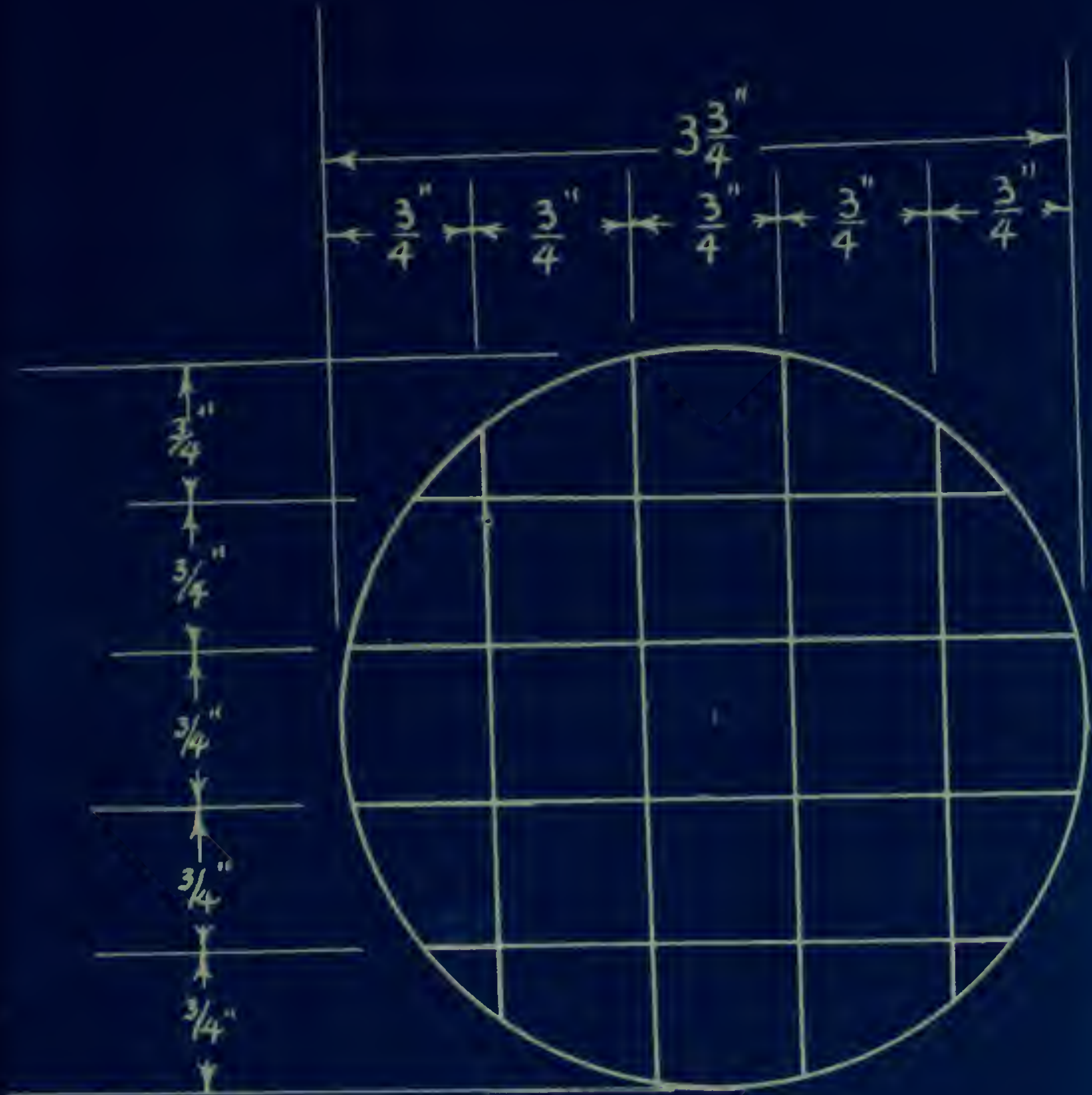
Weight, lb. per linear foot	-	0.49
***Ratio, Free to Face Area**	-	0.505

Center Line Spacing:

* Between Tubes	-	1.39 in.
* Between Rows	-	1.20 in.

* These dimensions apply to standard multi-row coil.


TOP VIEW
1'2"
1'2"
1'4"
1'2"
1'2"
Pc. 3
Pc. 2
Pc. 1
2"
Pc. 5
Pc. 4
3'4"
Pc. 6
END VIEWS
Pc. 4
Pc. 3
Pc. 2
Pc. 1
3 3/4"
5 3/16"
10 1/2"
3/4"
2"
3/4"
10 1/2"
3/4"
2"
3/4"
12"
12"
10 1/2"
NOTES:
1. END VIEW DIMENSIONS ARE INSIDE.
2. MATERIAL - STEEL 16 GAUGE.
3. FLANGED JOINTS TO BE MADE UP WITH 1/8" STD. GASKET MATERIAL AND BOLTED.
4. ONE SIDE PLATE OF PIECE 2 TO BE REMOVABLE. NUTS TO BE SECURED INSIDE DUCT.
5. DETAILS OF EGG CRATE FLOW STRAIGHTENER, PC. 5, ON PLAN NO. 2.
6. DETAILS OF NOZZLE, PC. 6, ON PLAN NO. 3.
7. PC. 7 TO BE MOUNTED IN PLACE OF REMOVABLE PLATE IN BOTTOM OF PC. 2.
SIDE VIEW
Pc. 2
Pc. 7 DRIP COLLECTOR
3/4"
5"
2 1/4"
1 1/2"
2 1/4"
5"
1'4"
PLAN NO. 1
DUCT WORK FOR TEST OF FINNED TUBE
THESIS - COURSE XIII A-6 MIT
R.E. JONES LT., U.S.N.
R.W. KING LT., U.S.N.
DEC. 28, 1948
R.W.K.



NOTES:

1. TO BE MOUNTED IN PC. 4 AS SHOWN ON PLAN NO. 1.
2. MATERIAL - STEEL 16 GAUGE OR LIGHTER.

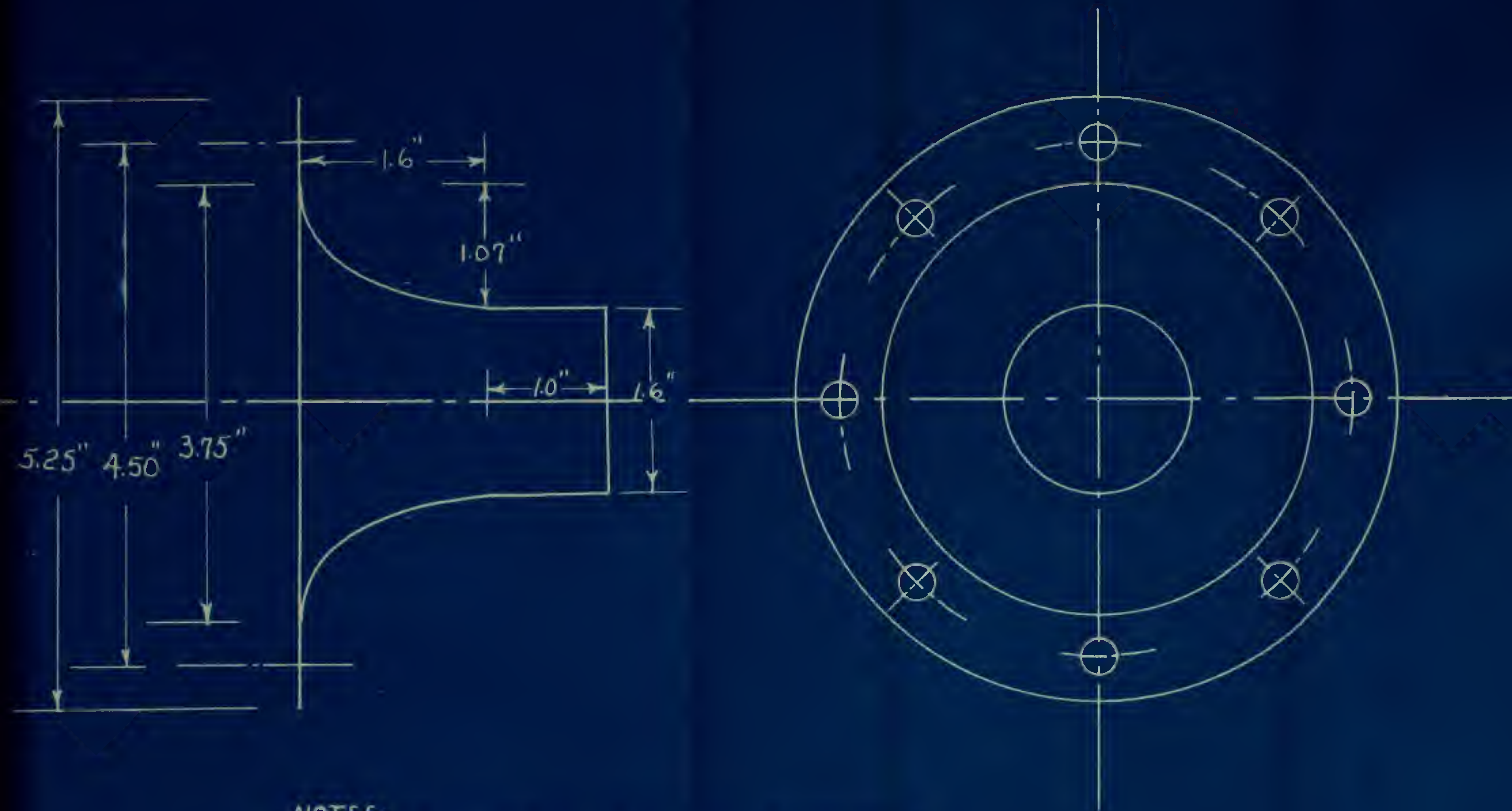
PLAN NO. 2

DETAILS PC. 5 — FLOW STRAIGHTENER

THESIS - COURSE XIII A-6 M.I.T.

R.E. JONES LT., U.S.N.   R.W. KING LT., U.S.N.

DEC. 29, 1948   R.W.K.



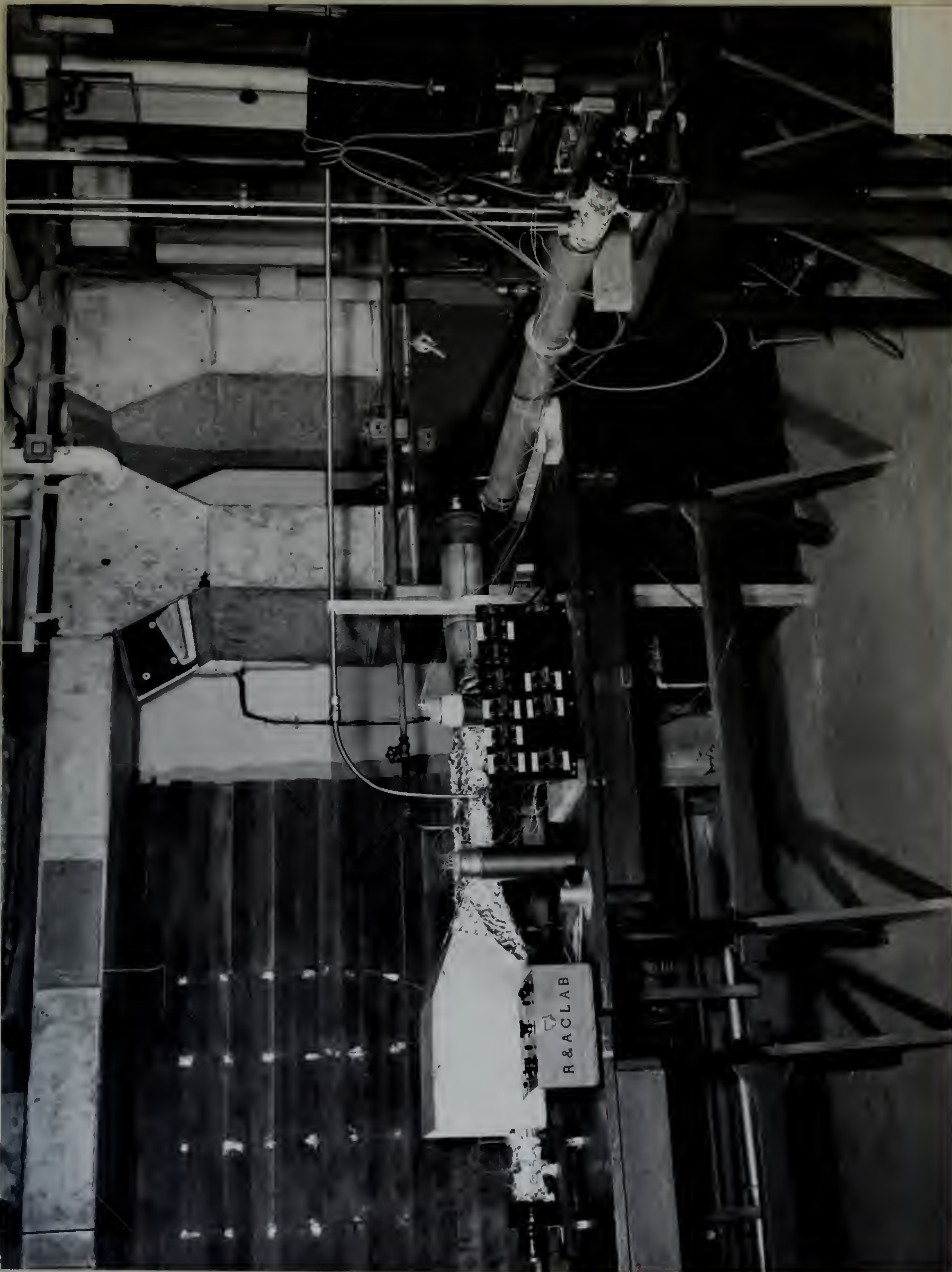
NOTES:

1. SPIN FROM 18 GAUGE ALUMINUM.
2. TO FIT FLANGE OF PC. 4 ON PLAN NO. 1.
3. DRILL FLANGE FOR $\frac{1}{4}$" BOLTS.

PLAN NO. 3
DETAILS OF MEASURING NOZZLE PC. 6
THESIS · COURSE XIII A-6 M.I.T.

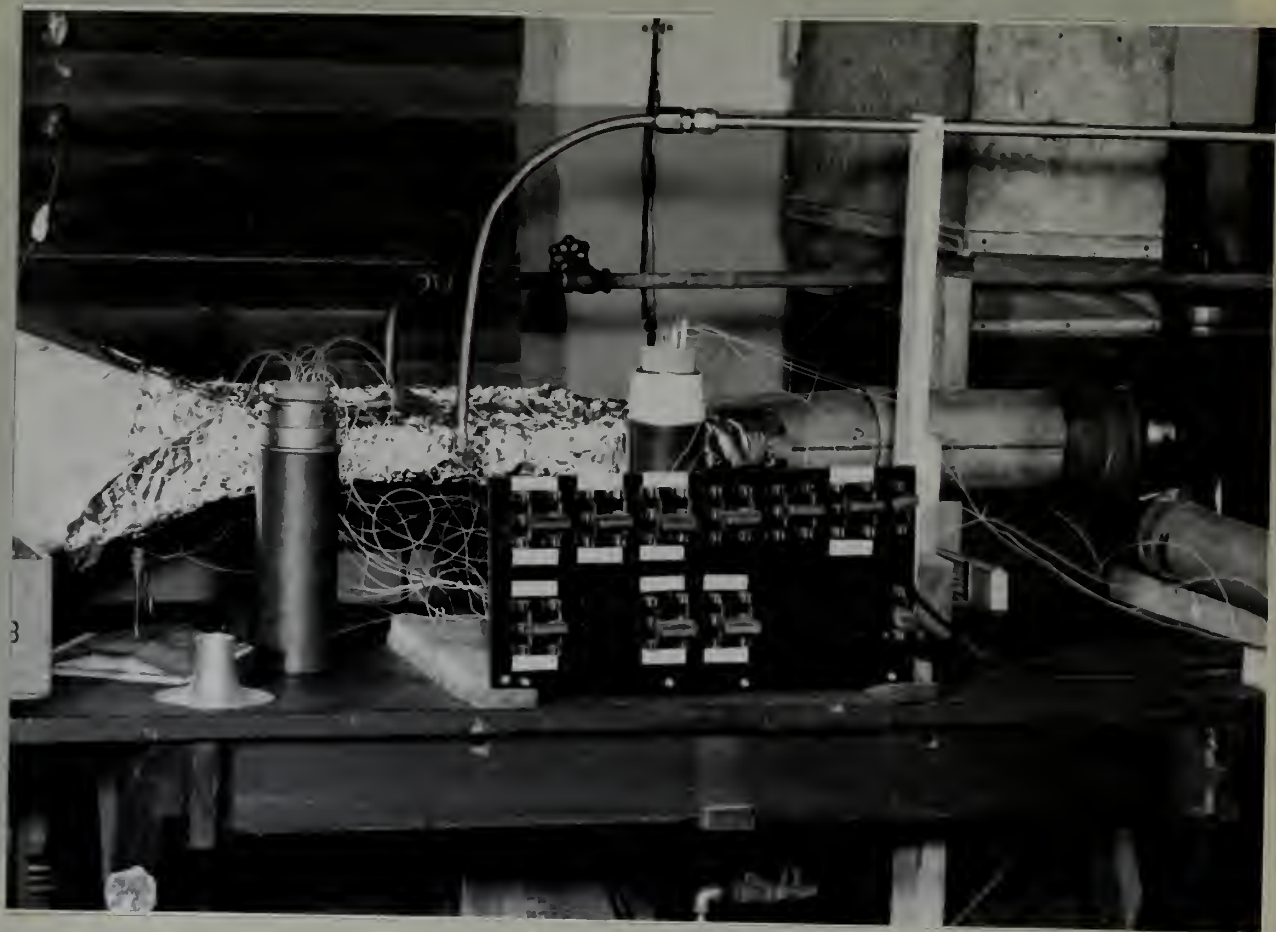
R.E. JONES LT., U.S.N — R.W. KING LT., U.S.N.
DEC. 29, 1948 — R.W.K.

FIGURE VIII



Arrangement of Equipment

FIGURE IX



Test Section

FIGURE X

Inlet End - Differential Thermocouples and Finned Tube

APPENDIX B

Notes on Procedure

1. A Leeds and Northrup indicating-type potentiometer was used for all thermocouple readings.

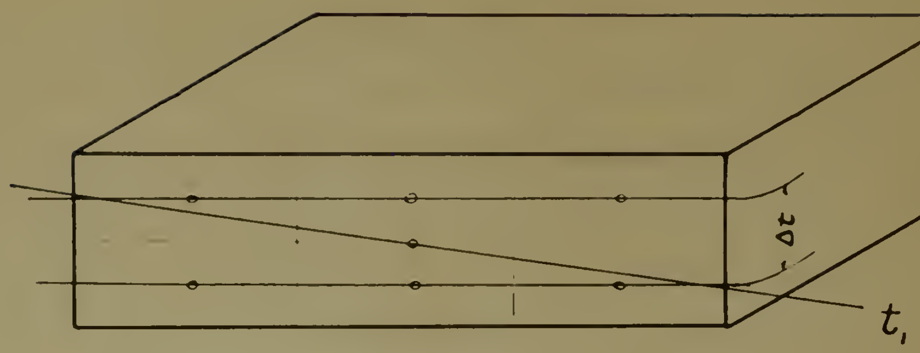
2. For runs 1 through 5 the following thermocouples were made up of No. 26 wire: t_1, t_1', t_1,Δt and t_c; all others were made of No. 30 wire. The remaining runs were made after the thermocouples measuring t_1 and t_1' were changed to No. 30 wire.

3. For all runs after No. 13, the drip pan and thermocouple measuring t_c were removed and the flat plate substituted.

4. Prior to run No. 14 the aspirating psychrometers were covered with aluminum foil in an attempt to bring humidity results into agreement. At the same time the pint thermos bottles, used for reference junctions, were changed to the quart size.

5. An air washer, with discharge into test room, was used for increasing the specific humidity of the inlet air for runs 10 through 13.

FIGURE XI



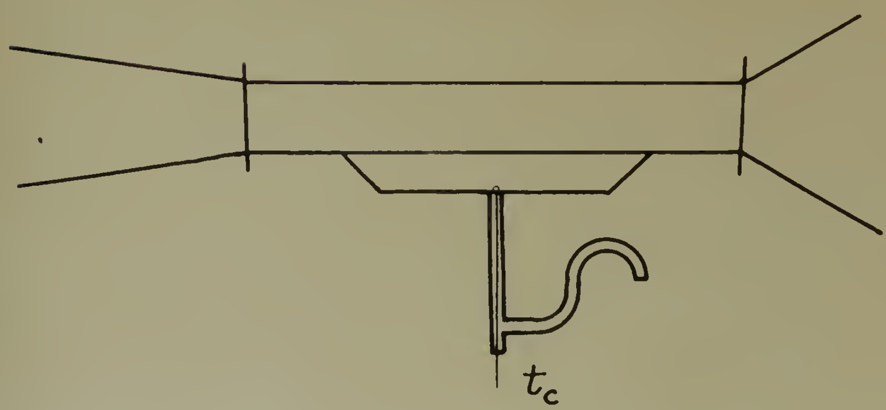


FIG XI TEST SECTION

SHOWING THERMOCOUPLE JUNCTIONS FOR MULTIPLE DIFFERENTIAL COUPLE MEASURING Δt AND FOR TEMPERATURES OF AIR INLET, t_1, AND CONDENSATE, t_c.

APPENDIX C

Summary of Data and Calculations

Table IV is the summary of the average of the data as observed in the fifteen thirty-minute runs. The thermocouple data were corrected by the deviation plots, Figures XII and XIII, to obtain the corrected data presented in Table V. The Notes on Procedure of Appendix B list the thermocouple wire used for each measurement.

The viscosity, specific volume and kinematic viscosity of water at low temperatures were obtained from data in Reference (4).

The Coefficient of Discharge curve, Figure XIV, is that for the standard nozzle in Reference (1).

TABLE IV
Average Data Uncorrected

Run No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Date	Mar.22	Mar.25	Mar.25	Mar.29	Mar.29	Mar.30	Mar.30	Mar.31	Mar.31	Apr.5	Apr.5	Apr.5	Apr.5	Apr.8	Apr.8
Duration	45 min.	45	30	30	30	30	30	30	15	30	30	30	30	30	30
(Pb)"Hg	30.29	30.35	30.35	30.12	30.12	30.00	30.00	29.88	29.88	30.12	30.12	30.12	30.10	29.57	29.57
(ti) m.v.	1.112	1.18	1.16	1.166	1.250	1.113	1.149	1.081	1.035	1.139	1.139	1.133	1.131	1.076	1.071
(ti') m.v.	0.755	0.66	0.66	0.603	0.666	0.579	0.603	0.483	0.480	0.903	0.903	0.930	0.923	0.604	0.603
(tl) m.v.	1.133	1.206	1.19	1.190	1.266	1.204	1.253	1.173	1.208	1.209	1.199	1.209	1.196	1.175	1.174
(Δt) m.v.	0.470	0.637	0.64	0.240	0.290	0.433	0.447	0.451	0.452	0.443	0.460	0.513	0.489	0.499	0.513
(tw1) m.v.	0.244	0.273	0.26	0.579	0.613	0.390	0.437	0.344	0.352	0.406	0.400	0.377	0.369	0.381	0.381
(tw2) m.v.	0.255	0.273	0.26	0.579	0.613	0.390	0.437	0.344	0.352	0.406	0.400	0.377	0.369	0.381	0.381
(to) m.v.	1.010	1.123	1.07	1.093	1.147	1.107	1.156	1.043	1.072	1.107	1.090	1.091	1.091	1.054	1.044
(to') m.v.	0.680	0.694	0.67	0.633	0.716	0.597	0.614	0.477	0.520	0.330	0.899	0.907	0.891	0.607	0.597
(tc) m.v.	--	--	--	--	--	--	--	--	--	--	1.054	1.053	1.034	--	--
Ww	4.93	19.5	18.89	1.525	1.30	6.73	3.39	10.98	10.47	13.06	12.86	32.06	31.02	14.22	14.08
(ts1) m.v.	0.340	0.331	0.30	0.656	0.701	0.447	0.554	0.391	0.400	0.480	0.449	0.419	0.406	0.443	0.430
(ts2) m.v.	0.381	0.346	0.33	0.730	0.769	0.494	0.616	0.417	0.440	0.519	0.501	0.444	0.440	0.483	0.470
(ts3) m.v.	0.377	0.346	0.33	0.730	0.769	0.494	0.616	0.417	0.440	0.510	0.504	0.443	0.437	0.460	0.454
(ts4) m.v.	0.592	0.595	0.57	0.371	0.913	0.714	0.806	0.641	0.670	0.626	0.623	0.534	0.591	0.680	0.660
(ts5) m.v.	0.383	0.367	0.34	0.751	0.787	0.516	0.650	0.431	0.447	0.521	0.507	0.444	0.446	0.437	0.471
(PN)" H_2O	0.476	0.406	0.337	0.556	0.444	0.612	0.438	0.423	0.618	0.608	0.434	0.444	0.614	0.579	0.421
(ΔPN)" H_2O	0.660	0.573	0.479	0.777	0.633	0.873	0.626	0.609	0.869	0.871	0.643	0.637	0.875	0.863	0.604

Note: Units are as indicated above for this data sheet only.

TABLE V
Corrected Data

Run No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Date	Mar.22	Mar.25	Mar.25	Mar.29	Mar.29	Mar.30	Mar.30	Mar.31	Mar.31	Apr.5	Apr.5	Apr.5	Apr.5	Apr.8	Apr.8
Pb	14.85	14.90	14.90	14.78	14.78	14.71	14.71	14.68	14.68	14.78	14.78	14.78	14.78	14.51	14.51
ti	79.8	82.8	81.7	82.1	85.6	83.0	84.6	81.5	81.7	84.1	84.1	83.9	83.8	81.3	81.1
ti'	64.8	61.4	61.4	58.3	60.8	58.9	59.8	54.5	54.4	73.6	73.6	74.8	74.5	60.0	60.0
t1	80.71	83.64	83.21	83.06	86.27	83.73	85.69	82.38	83.55	83.86	83.42	83.86	83.33	82.38	82.42
t2	77.09	79.18	78.55	81.31	84.16	80.57	82.47	79.07	80.25	80.62	80.04	80.13	79.68	78.73	78.67
Δ_t	3.62	4.46	4.66	1.75	2.11	3.16	3.22	3.31	3.30	3.24	3.38	3.73	3.65	3.65	3.75
tw1	43.32	44.89	44.17	59.34	60.44	50.22	54.68	48.05	48.41	50.93	50.66	49.62	49.24	49.80	49.80
tw2	43.83	44.89	44.17	59.34	60.44	50.22	54.68	48.05	48.41	50.93	50.66	49.62	49.24	49.80	49.80
to	78.4	83.4	81.0	82.1	84.5	82.7	84.9	79.8	81.2	82.7	81.9	82.0	82.0	79.4	79.9
to'	63.6	64.1	63.2	61.4	65.1	59.7	60.5	54.2	56.2	72.6	73.4	73.8	73.0	60.2	59.7
te	--	--	--	--	--	--	--	--	--	--	--	77.3	76.5	--	--
Ww	4.93	19.50	18.89	1.525	1.30	6.73	3.30	10.98	10.47	13.06	12.86	32.06	31.02	14.22	14.08
ts1	47.84	47.46	46.89	62.42	64.13	52.84	57.76	50.25	50.67	54.35	52.95	51.53	50.92	52.66	52.03
ts2	49.75	48.15	47.38	65.75	67.55	55.00	60.56	51.44	52.54	56.14	55.31	52.66	52.48	54.48	53.90
ts3	49.50	48.15	47.38	65.75	67.55	55.00	60.56	51.44	52.54	55.72	55.45	52.63	52.36	53.44	53.17
ts4	59.49	59.61	58.21	72.14	74.10	65.03	69.20	61.70	63.05	61.07	60.89	59.12	59.43	63.50	62.60
ts5	49.85	49.12	47.73	66.68	68.34	55.99	62.15	52.03	52.86	56.23	55.58	52.66	52.77	54.68	53.96
PN	0.0172	0.0146	0.0122	0.0200	0.0160	0.0221	0.0158	0.0153	0.0223	0.0219	0.0157	0.0160	0.0221	0.0216	0.0152
Δ_{pN}	0.0238	0.0207	0.0173	0.0281	0.0228	0.0315	0.0226	0.0219	0.0314	0.0314	0.0234	0.0230	0.0316	0.0311	0.0218
Σ_1	29.7	--	--	--	--	--	--	--	--	36.5	36.3	37.4	37.1	--	--
Σ_2	28.9	--	--	--	--	--	--	--	--	35.2	35.3	36.2	35.5	--	--

TABLE VI

Viscosity, Specific Volume and Kinematic Viscosity
of Water at Low Temperatures

Data from Reference (4), pp 407 and 413

t	μ'	v	ν
32	1.79	0.01602	19.27 X 10^{-6}
33	1.76	"	18.94 "
34	1.73	"	18.62 "
35	1.70	"	18.25 "
36	1.66	"	17.87 "
37	1.63	"	17.55 "
38	1.60	"	17.22 "
39	1.58	"	16.96 "
40	1.55	"	16.69 "
41	1.52	"	16.37 "
42	1.49	"	16.04 "
43	1.46	"	15.77 "
44	1.44	"	15.50 "
45	1.42	0.01603	15.29 "
46	1.40	"	15.08 "
47	1.38	"	14.81 "
48	1.35	"	14.54 "
49	1.33	"	14.33 "
50	1.31	"	14.11 "

μ' centipoises

v ft.^{3} lb.$^{-1}$

t degrees F

ν ft.^{2} sec.$^{-1}$

The viscosity in lbs. sec.$^{-1}$ ft.$^{-1}$ $= \mu' \times 6.72\ (10^{-4})$

$$\nu = 6.72\ (10^{-4})\ \mu'\ v$$

FIGURE XII

DEVIATION CURVE NO. 26 COPPER-CONSTANTAN WIRE

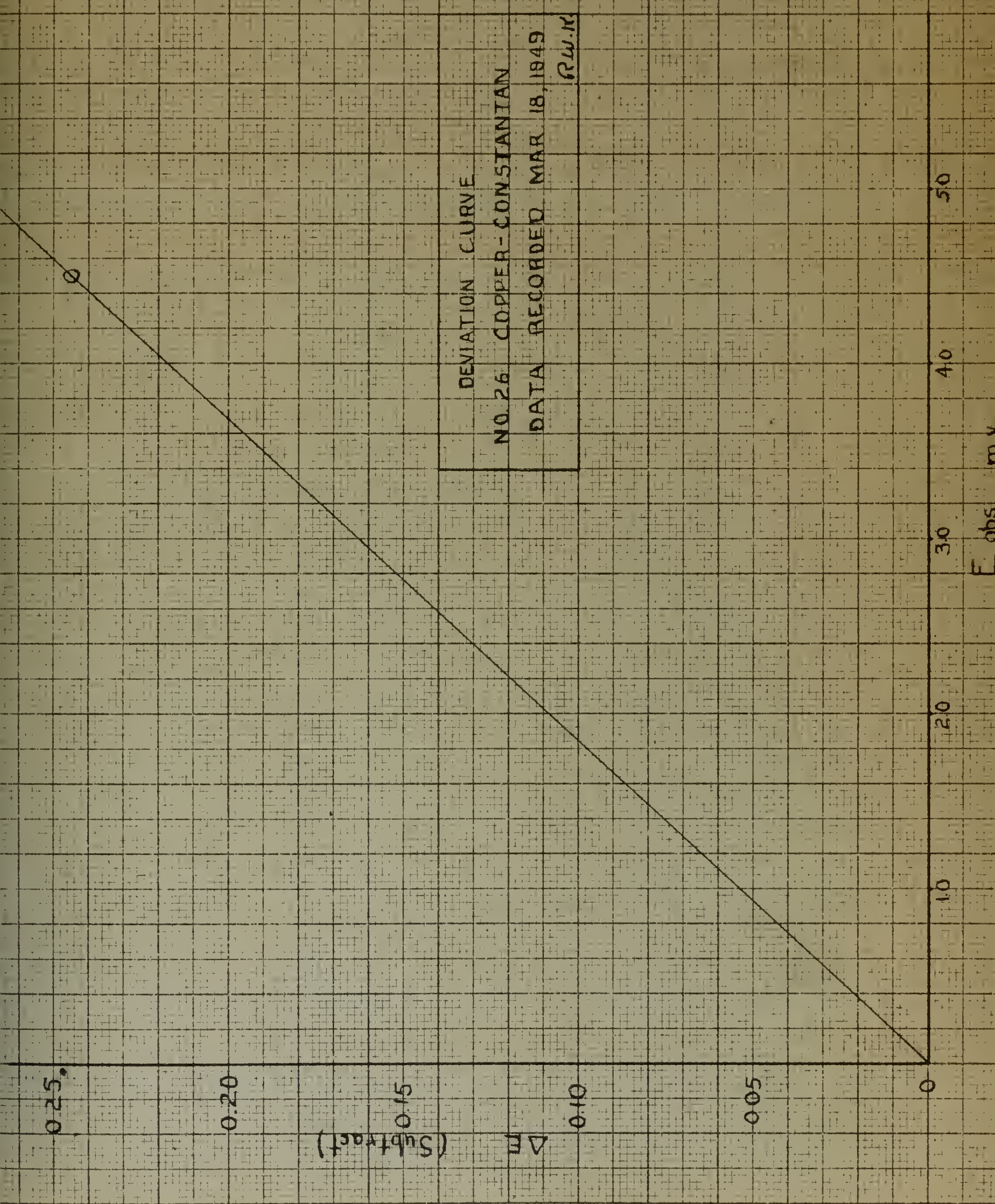
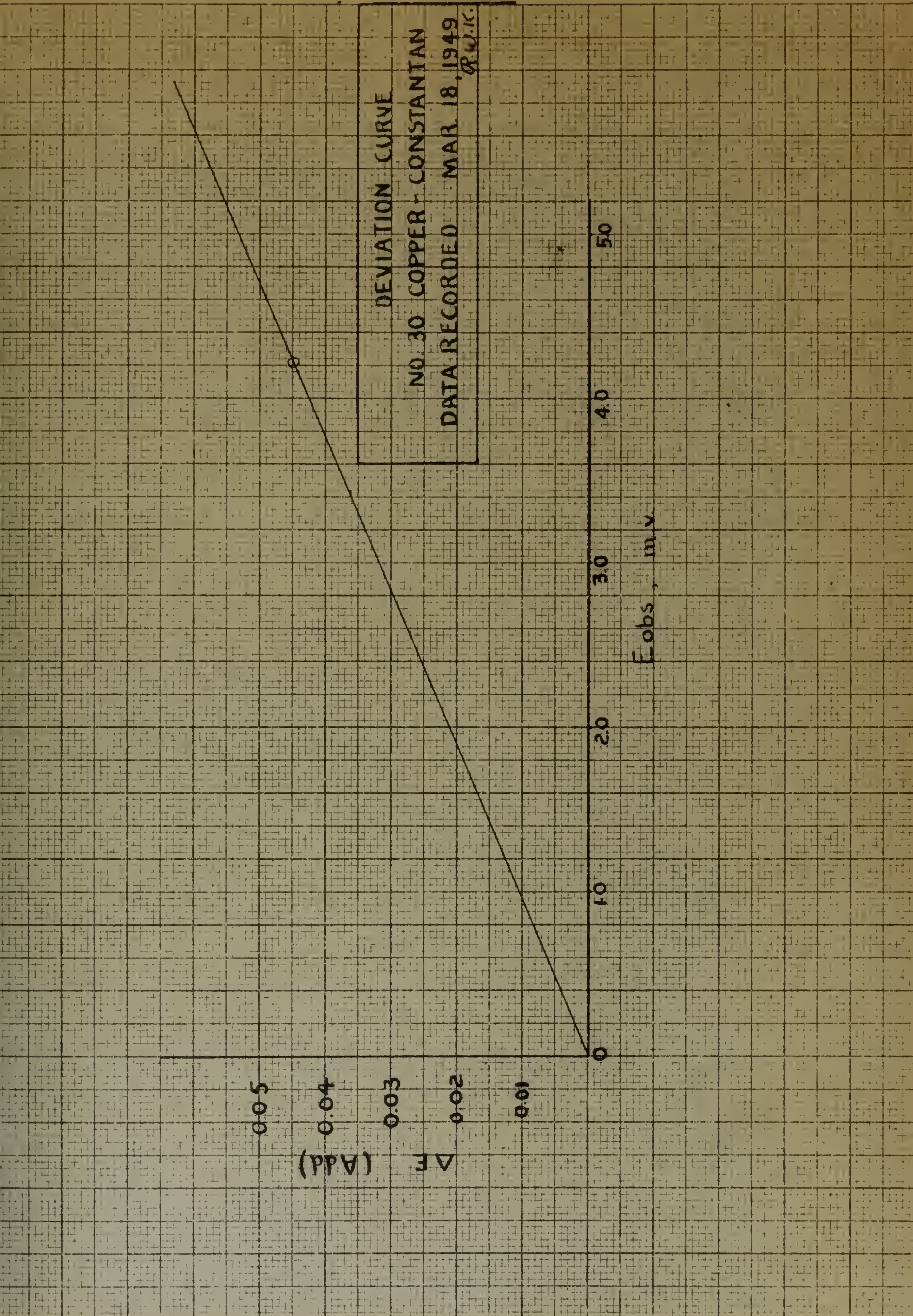
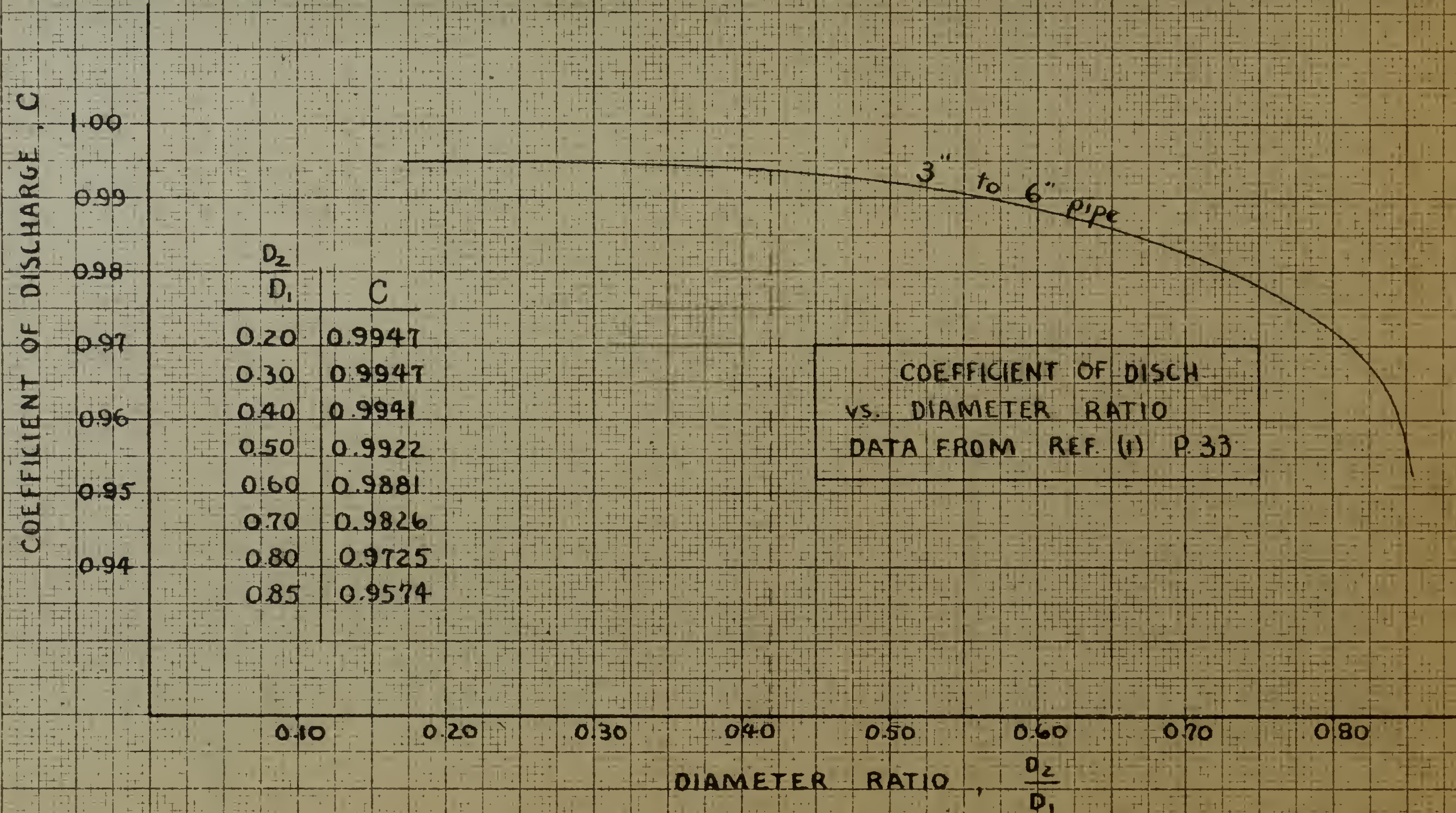


FIGURE XIII



DEVIATION CURVE
NO. 30 COPPER - CONSTANTAN
DATA RECORDED MAR. 18, 1949
R.W.K.
E obs, m.v.
0
1.0
2.0
3.0
4.0
5.0
ΔE (Add)
0.01
0.02
0.03
0.04
0.05

FIGURE XIV



$\frac{D_2}{D_1}$	C
0.20	0.9947
0.30	0.9947
0.40	0.9941
0.50	0.9922
0.60	0.9881
0.70	0.9826
0.80	0.9725
0.85	0.9574

APPENDIX D

Sample Calculations

Table VII shows the formulas used in the Sample Calculations. Formula (6) was obtained from Reference (7), Formula (19) from Reference (5), and Formula (20) from Reference (6). The remaining formulas are presented in any standard text of air conditioning heat transfer theory.

TABLE VII

FORMULAS USED IN COMPUTATIONS

1. $$W_2 = 0.622 \frac{P_{SN}}{P_{aN}} = 0.622 \frac{P_{SN}}{(P_b + P_N) - P_{SN}}$$

$$P_{SN} = \frac{P_b + P_N}{\left(\frac{0.622}{W_2} + 1\right)}$$

2. $$P_{aN} = (P_b + P_N) - P_{SN}$$

3. $$v_{aN} = \frac{R_a T_N}{P_{aN}\,(144)}$$

4. $$v_{mN} = \frac{v_{aN}}{1 + W_2}$$

5. $$\gamma_m = \gamma_s + \gamma_a$$

6. $$Q_m = 60\,(8.02)\,A_2 C \sqrt{\frac{\Delta P_N \times v_{mN} \times 144}{1 - \left(\frac{A_2}{A_1}\right)^2 \left[1 - 1.428 \frac{\Delta P_N}{(P_b + P_N)}\right]}}$$

7. $$Q_m = Q_a = w_m v_{mN} = w_a v_{aN}$$

8. $$Q_{as} = \frac{\gamma_a Q_a}{\gamma_{std.}}$$

9. $$V_{fs} = \frac{Q_{as}}{A_f}$$

10. $$t_2 = t_1 + \Delta t$$

11. $$c_{pm} = c_{pa} + W c_{ps} = 0.24 + W\,(0.45)$$

12. $$q_a = w_a\,(60)\left[(h_{a1} - h_{a2}) + (W_1 - W_2)\,h_{fc}\right]$$

$$q_a \cong w_a\,(60)\,c_{pm}\,\Delta t$$

13. $$q_w = w_w (60)(h_{f2} - h_{f1})$$

$$q_w \cong w_w (60)(t_{w2} - t_{w1})$$

14. $$V_w = \frac{w_w}{60\,(62.4)\,A_i}$$

15. $$R_e = \frac{V_w\,(D_i)}{\nu}$$

16. $$(t_s - t_w)_m = \frac{(t_{s1} - t_{w1}) - (t_{s5} - t_{w2})}{\ln\left(\frac{t_{s1} - t_{w1}}{t_{s5} - t_{w2}}\right)}$$

$$(t_s - t_w)_m \cong \frac{(t_{s1} - t_{w1}) + (t_{s5} - t_{w2})}{2}$$

17. $$q_a = \frac{(f_R)_o}{B} S_o\,(t_s - t_w)_m$$

$$(f_R)_o = \frac{B q_a}{S_o\,(t_s - t_w)_m}$$

18. $$t_{w\,av.} = \frac{t_{w1} + t_{w2}}{2}$$

19. $$(f_R)_c = 1.5\left[t_{w\,av.} + 100\right] \frac{(V_w)^{0.8}}{(12\,D_i)^{0.2}}$$ Ref. (5)

For $Re > 2100$

20. $$\%\ \text{sensible heat} = \frac{(t_1 - t_2)\,0.241 \times 100}{\text{total heat at } t_1' - \text{total heat at } t_2'}$$

$$\%\ \text{sensible heat} \cong \frac{(t_1 - t_2)\,0.241}{\Sigma_1 - \Sigma_2} \times 100$$

SAMPLE COMPUTATIONS

$$D_1 = \frac{3.72}{12} = 0.310 \text{ ft.}$$

$$D_2 = \frac{1.56}{12} = 0.130 \text{ ft.}$$

$$\frac{D_2}{D_1} = \frac{0.130}{0.310} = 0.419$$

$$A_1 = \frac{\pi (0.310)^2}{4} = 0.0755 \text{ sq. ft.}$$

$$A_2 = \frac{\pi (0.130)^2}{4} = 0.01327 \text{ sq. ft.}$$

$$\frac{A_2}{A_1} = \frac{0.01327}{0.0755} = 0.17586$$

$$\left(\frac{A_2}{A_1}\right)^2 = (0.17586)^2 = 0.0309$$

$$A_f = \frac{2.0 \times 10.37}{144} = 0.1440 \text{ sq. ft.}$$

$$S_o = 1.50 \frac{10.37}{12} = 1.295 \text{ sq. ft.}$$

$$S_i = 0.1505 \frac{10.37}{12} = 0.130 \text{ sq. ft.}$$

Run I:

$\underline{P_{SN}}$ $\quad P_{SN} = \dfrac{P_b + P_N}{\left(\dfrac{0.622}{W_2} + 1\right)} = \dfrac{14.85 + 0.02}{\dfrac{0.622}{0.0091} + 1} = \dfrac{14.87}{69.5} = 0.214 \text{ p.s.i}$

$\underline{P_{aN}}$ $\quad P_{aN} = (P_b + P_N) - P_{SN} = (14.85 + 0.02) - (0.214)$

$$P_{aN} = 14.66 \text{ p.s.i.a.}$$

$\underline{v_{aN}}$ $\quad v_{aN} = \dfrac{R_a T_N}{P_{aN}(144)} = \dfrac{(53.35)(538)}{(14.66)(144)} = 13.62 \; \dfrac{\text{cu. ft. dry air}}{\text{lb.}}$

$\underline{v_{mN}}$ $\quad v_{mN} = \dfrac{v_{aN}}{1 + W_2} = \dfrac{13.62}{1 + 0.0091} = 13.51 \; \dfrac{\text{cu. ft. mixture}}{\text{lb.}}$

$\underline{Q_m = Q_a}$ $\quad Q_m = 60(8.02)A_2 C \sqrt{\dfrac{\Delta P_N \times v_{mN} \times 144}{1 - \left(\dfrac{A_2}{A_1}\right)^2 \left[1 - 1.428 \dfrac{\Delta P_N}{(P_b + P_N)}\right]}}$

$$= 60(8.02)(0.0133)(0.993)\sqrt{\frac{(0.0238)(144)(13.51)}{1 - (0.0309)\left[1 - 1.428 \frac{(.0238)}{(14.87)}\right]}}$$

$$= 6.35\sqrt{\frac{46.4}{1 - (.0309)(0.998}} = 6.35\sqrt{\frac{46.4}{0.969}}$$

$$= 6.35(6.93) = 43.9 \; \frac{\text{cu. ft.}}{\text{min.}}$$

$\underline{w_a}$ $\quad w_a = \dfrac{Q_a}{v_{aN}} = \dfrac{43.9}{13.62} = 3.23 \; \dfrac{\text{lbs}}{\text{min.}}$

$\underline{w_m}$ $\quad w_m = \dfrac{Q_m}{v_{mN}} = \dfrac{43.9}{13.51} = 3.25 \; \dfrac{\text{lbs.}}{\text{min.}}$

$\underline{Q_{as}}$ $$Q_{as} = \frac{\gamma_a Q_a}{\gamma_{std}} = \frac{v_{std}}{v_{aN}} Q_a = w_a v_{std}$$

$$= \frac{13.37}{13.62}(43.9) = 43.1 \frac{\text{cu. ft.}}{\text{min.}}$$

$\underline{V_{fs}}$ $$V_{fs} = \frac{Q_{as}}{A_f} = \frac{43.1}{0.1440} = 299 \frac{\text{ft.}}{\text{min.}}$$

$\underline{c_{pm}}$ $$c_{pm} = c_{pa} + W c_{ps} = 0.24 + \frac{0.0097 + 0.0091}{2}(0.45)$$

$$= 0.24 + 0.0094(0.45) = 0.244$$

$\underline{q_a}$ $$q_a \cong w_a c_{pm} \Delta t \quad (60)$$

$$= (3.23)(0.244)(3.62)(60) = 171.0 \frac{\text{Btu}}{\text{hr.}}$$

$\underline{V_w}$ $$V_w = \frac{w_w}{60(62.4)A_i} = \frac{4.93}{60(62.4)\left(\frac{0.260}{144}\right)} = 0.73 \frac{\text{ft.}}{\text{sec.}}$$

$\underline{Re}$ $$Re = \frac{V_w (D_i)}{\nu} = \frac{(0.73)\left(\frac{0.575}{12}\right)}{15.7 \times 10^{-6}} = 2230$$

$\underline{(t_s - t_w)_m}$ $$(t_s - t_w)_m \cong \frac{(t_{s1} - t_{w1}) + (t_{s5} - t_{w2})}{2}$$

$$= \frac{(47.84 - 43.32) + (49.86 - 43.83)}{2} = 5.27^\circ F$$

$\underline{(f_R)_o}$ $$(f_R)_o = \frac{B q_a}{S_o (t_s - t_w)_m}$$

$$= \frac{10\,(171)}{1.5\left(\frac{10.37}{12}\right)(5.27)} = 251 \frac{\text{Btu}}{\text{hr. - sq. ft - }^\circ\text{F}}$$

$\underline{t_{wav}}$ $$t_{wav} = \frac{t_{w1} + t_{w2}}{2} = \frac{43.32 + 43.83}{2} = 43.57^\circ F$$

$\underline{(f_R)_c}$ $$(f_R)_c = 1.5\left[t_{wav} + 100\right] \frac{V_w^{0.8}}{(12 D_i)^{0.2}}$$

$$= 1.5\left[43.57 + 100\right] \frac{(0.73)^{0.8}}{(0.575)^{0.2}}$$

$$= 1.5\left[143.6\right] \frac{0.778}{0.895} = 187 \frac{Btu}{hr\text{-}sq.ft.\text{-}^\circ F}$$

<u>% sensible heat</u> $$\%\ \text{sensible heat} \cong \frac{(t_1 - t_2)\,0.241}{\Sigma_1 - \Sigma_2} \times 100$$

$$= \frac{(3.62)(0.241)}{(29.7 - 28.3)} \times 100$$

$$= 62.4\%$$

APPENDIX E

Nozzle Calibration

In Chapter V the need for an accurate determination of air flow rate is discussed. Reference (7) and its appendixes give pertinent information on the choice of a flow measuring device for air and its method of calibration. The authors of reference (7) found values of the coefficient of discharge higher than those given in Figure XIV. Although their results would not affect the experimental results shown in Chapter IV to any great extent it is desirable that a calibration curve for the nozzle installed in the test equipment be obtained for any future work on the equipment. For this reason the method of calibration is described in detail in the following paragraphs.

The correct flow rate is assumed to be that resulting from the mean value of the velocity pressure as obtained by pitot tube traverse. Figure XV shows the locations and values of r (distance from the center of the duct to the traverse point) for an eight point traverse.

A vertical and horizontal traverse is made and the readings averaged for the four areas. The mean value of $\sqrt{h}$ for the full nozzle $= \dfrac{\sqrt{h_1}+\sqrt{h_2}+\sqrt{h_3}+\sqrt{h_4}}{4}$ where h_1, h_2, h_3, and h_4 are respectively the average readings in inches of water of the velocity pressure of areas 1, 2, 3, and 4.

The following formulas give the accurate flow rate:

$$V = 4006\sqrt{\frac{h}{x}}$$

$$Q_p = V \cdot (\text{cross-section area of duct})$$

V Velocity, feet per minute.

h Velocity head, inches of water at 70 degrees F.

x Density of actual air relative to air at 70 degrees F. and 29.92 inches of mercury.

Q_p Flow rate as measured by pitot tube traverse, cubic feet per minute.

The pitot traverse must be installed as far upstream from the nozzle as possible to cause a minimum disturbance of flow at the nozzle inlet. In test equipment used in this report, six to eight inches will probably give accurate results provided that a small pitot or impact tube is used.

For each flow rate the pressure drop across the nozzle and the static pressure upstream of the nozzle are measured. The quantity of air flowing through the nozzle as measured by the pressure drop is given by the following formula:

$$Q = 8.02\, A_2 \sqrt{\frac{\Delta p \times v_1}{1-\left(\frac{A_1}{A_2}\right)^2\left(1-1.428\,\frac{\Delta p}{p_1}\right)}}$$

A_2 Nozzle throat area, sq. ft.

A_1 Duct area, sq. ft.

Δ_p Pressure drop from point p_1 above the nozzle to point p_2 following the nozzle, in pounds per square foot.

v_1 Specific volume of air at nozzle entrance.

Q Flow rate, cubic feet per second.

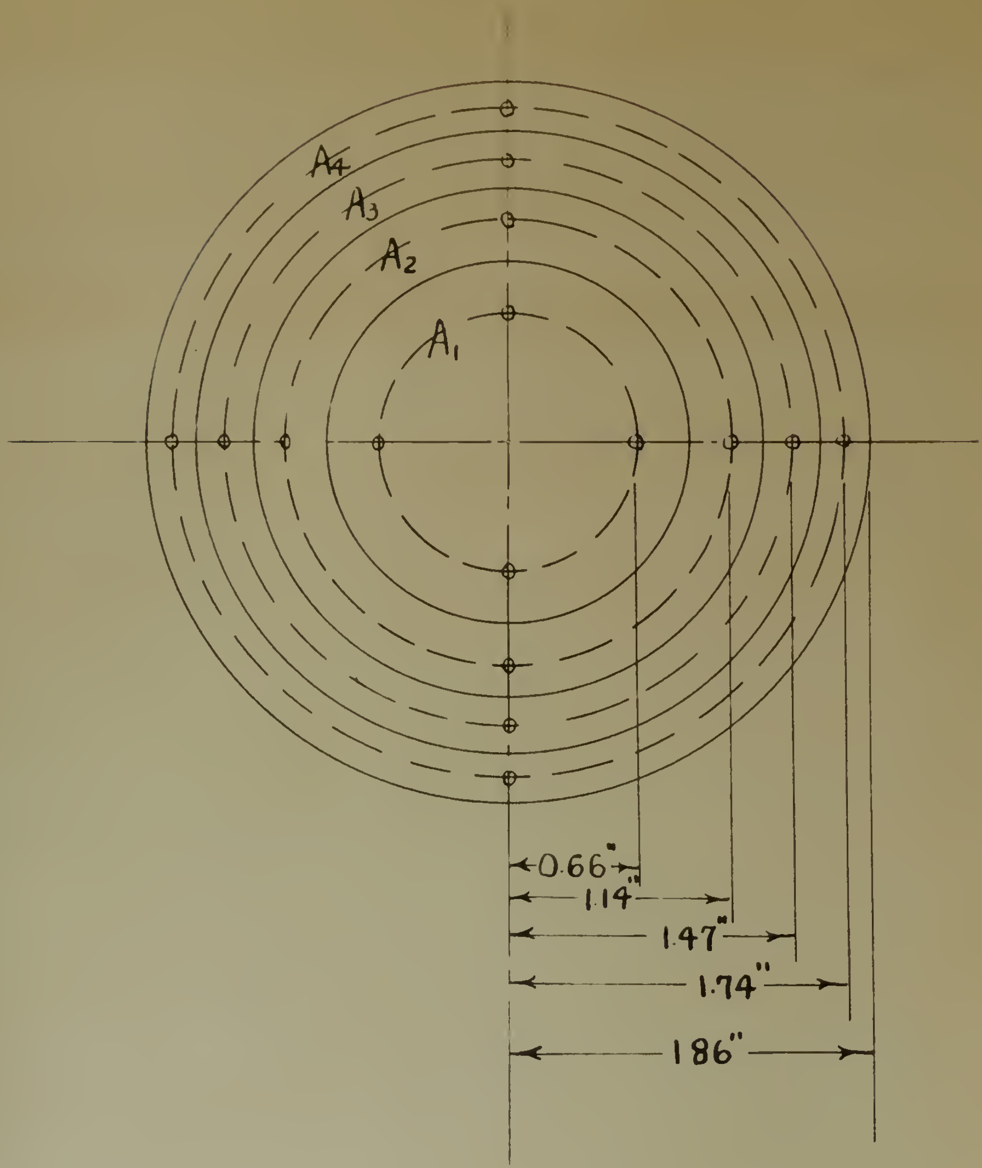
$Q_N = 60Q$ Flow rate as measured by nozzle pressure drop, cubic feet per minute

The Nozzle Coefficient of Discharge = $C = \dfrac{Q_P}{Q_N}$

Data from several runs using the above procedure can be used to plot a curve of the coefficient of discharge versus the pressure drop across the nozzle (C vs Δp). Then the pitot traverse can be removed and the flow rate can be determined accurately by the product of Q_N and C.

FIGURE XV

LOCATION OF TRAVERSE POINTS



A_1, A_2, A_3 and A_4 are concentric equal areas.

Dash lines divide each area into two equal areas.

Graphical Method of Determining the Specific Volume of Dry Air

In determining coil characteristics all values are based on the rate of flow of dry air. The specific volume of dry air is a relatively easy quantity to calculate but involves equations (1), (2) and (3) as shown in the Sample Calculations, Appendix D.

As a result of the steps necessary to determine the dry air specific volume, Fig. XVI was developed. The graphical method uses information which is determined from instrument readings: the barometric and duct static pressures, wet and dry bulb temperatures for specific humidity and dry bulb temperature for a small correction factor since the primary graph is made up for 70 degrees F. The derivation of the expression for the specific volume of dry air involving the different variables follows:

$$P_{aN} = (P_b + P_N) - P_{sN} \qquad (2)$$

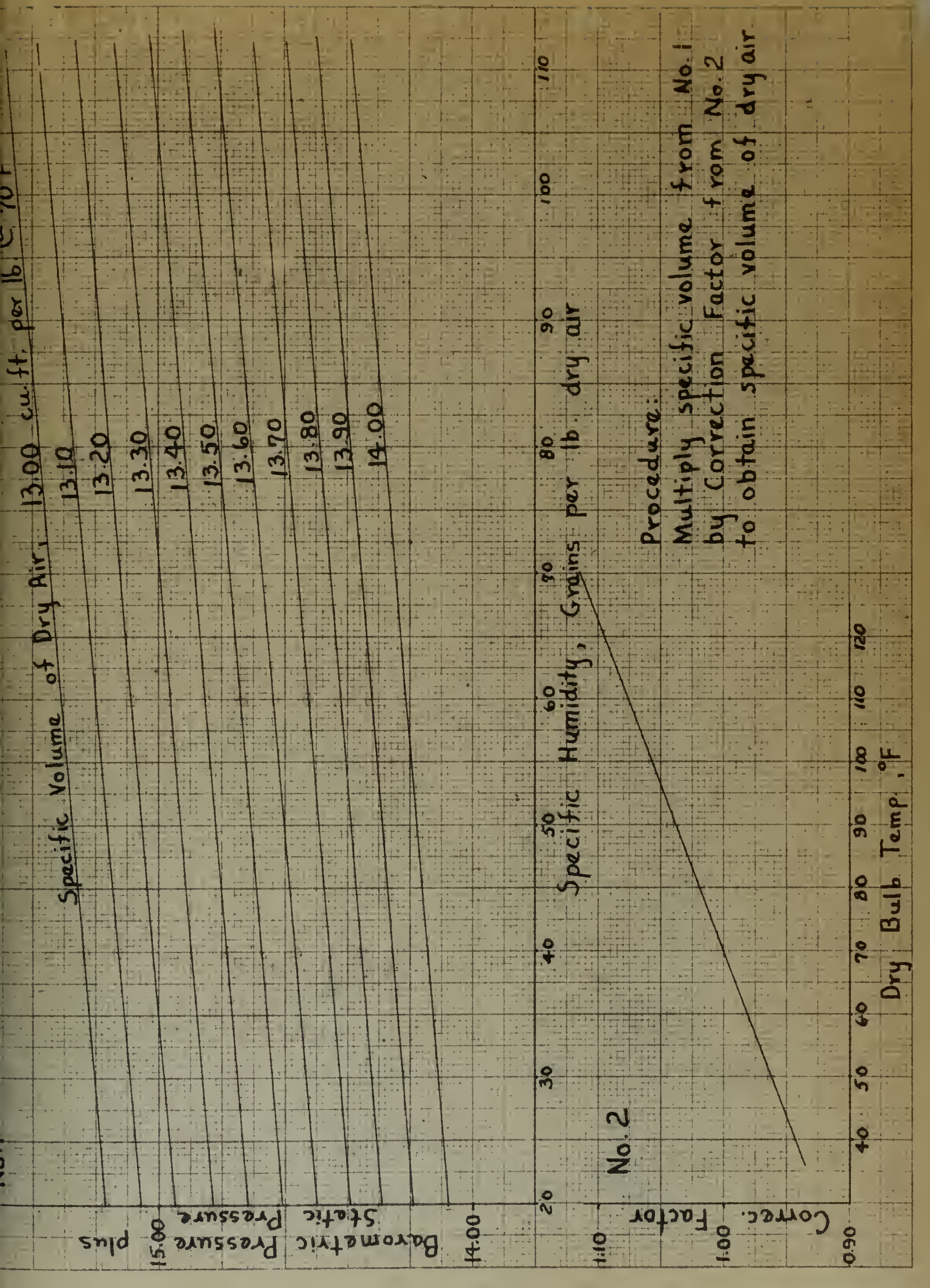
$$P_{sN} = \frac{(P_b + P_N)}{\left(\frac{0.622}{W_2} + 1\right)} \qquad (1)$$

$$P_{aN} = \frac{R_a T_N}{144\, v_{aN}} \qquad (3)$$

$$\frac{R_a T_N}{144\, v_{aN}} = (P_b + P_N) - \frac{(P_b + P_N)}{\left(\frac{0.622}{W_2} + 1\right)}$$

$$v_{aN} = \frac{R_a T_N}{144\,(P_b + P_N)\left[1 - \frac{1}{\left(\frac{0.622}{W_2} + 1\right)}\right]} = \frac{R_a T_N}{144\,(P_b + P_N)}\left(1 + \frac{W_2}{0.622}\right)$$

FIGURE XVI



Specific Volume of Dry Air, 13.00 cu.ft. per lb.
13.10
13.20
13.30
13.40
13.50
13.60
13.70
13.80
13.90
14.00
Specific Humidity, Grains per lb. dry air
20
30
40
50
60
70
80
90
100
110
Procedure:
Multiply specific volume from No. 1
by Correction Factor from No. 2
to obtain specific volume of dry air
No. 2
Dry Bulb Temp., °F
40
50
60
70
80
90
100
110
120
Barometric Pressure plus Static Pressure
15.00
14.00
Correc. Factor
1.10
1.00
0.90

APPENDIX F

Original Data

The data obtained in accordance with the procedure described in Chapter III of this report were recorded in a computation notebook. This notebook is in the possession of Professor A. L. Hesselschwerdt of the Mechanical Engineering Department.

Table IV, Appendix C, is a summary of the recorded data averages for all test runs.

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